UNIVERSITY OF ALBERTA

PERFORMANCE AND EMISSIONS OF A DUAL FUEL TWO STROKE BUS ENGINE

BY

© **PRZEMYSLAW PETER NOWAK**

A thesis submitted to the Faculty of Graduate Studies and Research in partial fulfillment of the requirements for the degree of **MASTER OF SCIENCE.**

DEPARTMENT OF MECHANICAL ENGINEERING

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Przemyslaw Nowak

Przemyslaw Peter Nowak
#2 11425 - 95 Street
Edmonton, Alberta
T5G 1L3

Date: October 6, 1994

UNIVERSITY OF ALBERTA

FACULTY OF GRADUATE STUDIES AND RESEARCH

The undersigned certify that they have read, and recommend to the Faculty of Graduate Studies and Research for acceptance, a thesis entitled **Performance and Emissions of a Dual Fuel Two Stroke Bus Engine** submitted by **Przemyslaw Peter Nowak** in partial fulfillment of the requirements for the degree of **Master of Science**.

Dr. M.D. Checkel (Supervisor)

J D Dale
Dr. J.D. Dale

Dr. L.W. Kostiuk

Dr. J.T. Ryan

Date: Oct 5 /94

ABSTRACT

In this study, a Detroit Diesel 6V-71 transit bus two stroke engine was converted to dual fuel natural gas/diesel operation. The Electronic Dual Fuel Injection System of Alternative Fuel Systems Inc. was used as the basis for the conversion. To make the conversion system simple and affordable, minimal changes were done to the stock engine. The existing compression ratio, blower drive, and diesel injection system were retained as much as possible. The existing diesel fuel injectors were used to inject pilot diesel as controlled by an electronic rack actuator. The gaseous fuel was delivered to each cylinder by individual high speed solenoid valves which controlled the timing and volume of injected natural gas. The gas valves were installed on the outside of the engine block and delivered gas through tubes to each cylinder air intake port. The converted engine retained the full capability to operate on straight diesel fuel.

The conversion kit was developed and optimized for this engine by preliminary testing. To compare performance and emissions, the engine was tested in both diesel only and dual fuel mode at maximum torque and over the Japanese 13-mode test. All tests were carried out on an eddy-current type dynamometer. On dual fuel, the engine performance and emissions were investigated for the two different sets of gas injection tubes. The first set of tubes was deflected downwards towards the piston while the second set of tubes was deflected upwards towards the top of the cylinder.

The converted engine maintained its original power and torque levels. The specific fuel consumption was higher in dual fuel modes than in straight diesel. This was mainly due to high natural gas losses encountered during the scavenging process. The high compression ratio (18.7:1) and high air intake charge temperatures limited the diesel

fuel replacement ratio. The maximum diesel replacement ratios of about 46 % were achieved at 800 engine rpm and maximum torque conditions. As expected for natural gas fuelling, the CO_2 emissions were reduced in dual fuel modes. On dual fuel, the composite CO, NMHC, and soot emissions were higher while the composite NO_x emissions were lower compared with diesel only results. The gas injection tubes pointed down performed better, producing significantly lower CO, NMHC, and soot emissions than tubes pointed up configuration.

The conversion system performed very well over the entire operating range of the 6V-71 engine and did not cause any technical problems throughout the testing. However, to optimize this engine for dual fuel operation, a reduced compression ratio and gas injection above the intake ports would be required.

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CHAPTER 1 INTRODUCTION

1.1 BACKGROUND TO THIS INVESTIGATION

Petroleum, since the turn of the century, has been an inexpensive and readily available energy source for internal combustion engines. Experts for some years now have predicted the end of the era [40]. The world's supply of crude oil is being depleted, creating incentives for promotion and further evaluation of alternative fuel sources. Moreover, considerable attention has been focussed on reducing exhaust emissions from motor vehicles in recent years. Much of this attention is because of the situation in many large urban centres in North America where air quality is having a measurable adverse effect on man, animals, and vegetation [11]. The anthropogenic (man-made) pollutants contribute mostly to atmospheric pollution. The emissions from internal combustion engines represent a major portion of the anthropogenic pollution generated in the world [11]. As a further complication, the undesirable emissions from transportation sources tend to be concentrated in urban areas where a large population is exposed to high levels of pollutant emissions and their effects can be more pronounced. In general, the future oil supply limit and the necessity to improve air quality call for efforts throughout all areas of transportation to develop any alternative fuel which is both abundant and clean burning.

The transition to non-petroleum vehicular fuels is underway. Although many alternatives such as alcohols, natural gas, and hydrogen fuels are being studied and developed, it appears, that natural gas can displace a significant fraction of petroleum transportation fuel in near future. Natural gas has been proposed as the most viable alternative fuel due to its several inherent advantages [24]. These include ease of distribution using the existing pipeline networks, low cost, low emissions, and relatively easy adaptability to current engines.

Canada has been a net importer of crude oil products since 1975 [3]. The cost of imported oil is relatively high and its supply is not secure [3, 13]. The substitution of low cost, abundant, domestically produced natural gas for high cost imported oil is an economic advantage to all Canadians. Natural gas is one of Canada's most abundant natural resources. The long term supply outlook for natural gas is much more secure than for either propane, gasoline or diesel fuel [3].

Table 1.1 shows Canada's crude oil and natural gas reserves, as well as the annual production and consumption rates for 1992. Alberta's production rates of crude oil and natural gas are also included in Table 1.1.

Table 1.1 Canada reserves, production, and consumption of crude oil and natural gas [10].

Fuel Type	Remaining Proven Reserves at 31.12.1992	Production Canada 1992	Production Alberta 1992	Consumption Canada 1992
Crude Oil	809,734,000 cubic metres	71,265,000 cubic metres	52,537,000 cubic metres	79,430,935 cubic metres
Natural Gas	2,671,554[1)]	119,000[1)]	97,824[1)]	56,173[1)]

[1)] Million cubic metres at 101.325 kPa and 15°C

Based on the 1992 statistics, the annual consumption of crude oil in Canada was 79.4 million cubic meters, about 11.5 % higher than the annual production. This means that

8.2 million cubic metres of crude oil had to be imported from foreign resources in 1992. At the same time, the annual consumption of natural gas in Canada, 56 billion cubic metres, was about 52.8 % lower than the annual production. Thus, Canada was able to export 62.8 billion cubic metres of natural gas in 1992. It is shown in Table 1.1 that the province of Alberta is the main producer of crude oil and natural gas in Canada. More than 82 % of the total Canadian production of natural gas came from the province of Alberta in 1992. In addition to the vast conventional reserves, natural gas is renewable and can be obtained from a wide variety of unconventional resources such as sewage, biomass, coal bed degasification as well as other new technologies [13]. Therefore, it is reasonable to expect that the natural gas production can be maintained at the same levels that are presently being produced.

Since vehicles are the prevailing source of atmospheric pollution in metropolitan areas, great attention must be paid to reducing emissions from mass transportation vehicles, particularly from transit buses. Currently, bus systems mainly operate on diesel fuel due to the good inherent fuel economy of the high compression diesel engine and its durability. There are, however, two major problems associated with diesel engined buses. One is the relatively high cost of diesel fuel and the other is pollution from diesel engines, especially particulate and nitrogen oxides emissions. The requirement for cleaner environment has been the driving force behind the evolution of more stringent emission legislation limits. In Canada, emissions regulations are set by both Environment Canada and Transport Canada. Their interest in urban bus pollution has increased during the last decade. Until 1985, urban buses had not been considered a serious environmental problem. They produced relatively low levels of hydrocarbon and carbon monoxide emissions and the number of buses was small compared with the total number of motor vehicles [17]. However, with tighter standards for passenger cars and trucks, emissions from buses became more important. According to EPA, public exposure to urban bus exhaust emissions is now much higher than exposure from many other pollution sources [17]. Consequently, in 1989, Environment Canada announced plans to have in place the most stringent regulations that diesel engine technology would allow. Currently the 1990 U.S. emissions standards are in effect in Canada. According to Transport Canada, commencing with the 1995 model year all heavy duty diesel engine powered vehicles marketed and sold in Canada will have to comply with the corresponding U.S. EPA exhaust emissions standards. The more stringent U.S. emission standards will be achievable only under the condition that low sulphur diesel fuel will be commercially available throughout Canada at that time. Table 1.2 presents the current and planned emission standards for urban buses powered by heavy-duty diesel engines in Canada and the United States.

Table 1.2 The current and planned urban bus heavy-duty engine exhaust emission standards in Canada and the United States [11, 26].

Country	HC (g/kW.h)	CO (g/kW.h)	NOx (g/kW.h)	Particulate Matter (g/kW.h)
Canada (1990)	1.7	20.8	8.0	0.8
U.S.A (1994-1997)	1.7	20.8	6.7	0.094

Note: Measured during the EPA transient test for heavy-duty diesel engines.

With the planned stringent exhaust emission standards, any technology that may reduce the exhaust pollutants is worth of evaluation. One potential method for improving urban gas exhaust emissions is the use of a cleaner burning alternative fuel and in particular natural gas. The conversion of the heavy duty diesel engines to dual fuel operation is being considered as a viable method to reduce exhaust emissions. The dual fuel concept involves the utilization of the gaseous fuel by mixing it with the intake air, and ignition of the premixed charge is accomplished by injecting a fraction of liquid diesel fuel near top dead centre of the compression stroke.

Existing buses not only contribute to an air pollution problem in many cities; they also consume large amounts of diesel fuel, which could be replaced by natural gas. The energy cost of natural gas is lower than the energy cost of diesel fuel [1, 17]. Because of diminishing reserves of crude oil in the world, one can expect a continual increase in the cost of diesel fuel in the near future. Since most of the transit buses are powered by diesel engines, the cost of diesel fuel becomes a large fraction of the total operating expenditures. Table 1.3 illustrates the total distance travelled, the total number of passengers carried, and the total diesel fuel consumption by the transit buses in the province of Alberta and in Canada in 1993.

There are, however, associated problems with the use of natural gas. Since the volumetric energy content of compressed natural gas (CNG) is about five times lower than for diesel fuel, the travelling range of the vehicle is significantly affected. Consequently, the storage space and weight of the natural gas cylinders are another problem connected with using natural gas. Additional obstacle is a high cost of compressor stations, risk of failure of single source of fuel (i.e. the compressor), and explosion hazards for indoor storage. Moreover, there is still an insufficient number of accessible, high capacity, fast-fill natural gas fuelling stations in Canada. Also refuelling time is greater with inherent cost to operators.

Table 1.3 Urban transit statistics for Alberta and Canada - 1993 [9].

	Alberta	Canada
Carriers Reporting (No.)	8	82
Total Distance Run	60,782,040 km	497,298,242 km
Total Passengers Carried	105,752,916	911,749,453
Diesel Fuel Consumed	33,283,332 litres	319,964,237 litres

Considering the advantages and disadvantages coupled with the use of natural gas, large users, such as urban transit fleets, seem to be the best potential markets for natural gas vehicles. Most of them have their own fuelling stations. Urban transit fleets make heavy use of vehicles, mainly within the metropolitan area, which allows them to maximize natural gas consumption and to achieve large annual fuel cost savings due to the high volume of diesel fuel consumption displaced. For fleets, such as city buses, where the vehicles travel only limited distances and return to the depot on regular basis, the limit on range and the space required for storage cylinders have a small effect. The added weight of storage cylinders does not constitute a problem for the robust transit buses. For roadway protection, however, the total mass of transit bus with the storage cylinders should not exceed the relevant allowable weight limits.

Natural gas is a very attractive fuel from an economical and an environmental points of view and is especially suited to fleet vehicles. In addition, the use of natural gas as a vehicular fuel can play an important role in helping to strengthen national energy security. The possibility of substituting cleaner-burning fuel for diesel fuel in automotive service has recently drawn increasing attention. Although the technology for conversion of gasoline-engined cars and light trucks to natural gas operation is already quite well established, the operation of diesel fuelled heavy trucks and buses on natural gas is still not well commercialized. Thus, the development of a dual-fuel natural gas/diesel engine is a very important task because of the potential for substituting a cleaner, less expensive, and more plentiful fuel for diesel fuel.

1.2 OBJECTIVES OF THIS INVESTIGATION

Existing diesel bus engines will have difficulty meeting stringent emission standards in the future. About 85-90 % of all diesel buses currently used in Canada are powered by 6V-71 two-stroke engines manufactured by the Detroit Diesel Corporation (DDC) [11]. However, the 6V-71 engine is no longer available in a configuration which meets the current emissions standards presented in Table 1.2. The new exhaust emissions standards promulgated for heavy duty diesel engines forced the transit bus operators to start switching from DDC 6V-71 engine to the cleaner DDC 6V-92TA turbocharged and aftercooled engine. However, the replacement of current 6V-71 engined buses with cleaner new-generation buses would take approximately 16-20 years at normal fleet renewal levels. Converting some of these existing vehicles to alternative fuels is another possibility which has to be considered. Natural gas has been recommended as the most practical alternative compression-ignition engine fuel due to its several natural advantages such as low cost, availability, low pollutant emissions, and relatively easy adaptability to diesel engines [2, 19, 40].

The choice to retrofit existing engines to dual fuel natural gas/diesel operation is a fast and a low cost solution. The purchase of new engines is much more expensive than the conversion of existing engines to dual fuel operation [11]. By converting existing buses as well as buying new natural gas buses, the transit bus fleets could take advantage of large natural gas fuel use. However, dual fuel natural gas/diesel engines still receive little research and development attention. Many conversion systems do not take full advantage of the possibilities the dual fuel engine could offer. Because the conversion of a four stroke type diesel engine is much less demanding, most of the dual fuel engine research in the past was conducted on this type of engine. Therefore, the conversion of the 6V-71 two stroke diesel engine became a very challenging and high priority task.

The main objectives of this study were:

1. Develop a prototype conversion kit (hardware and software) for the Detroit Diesel 6V-71 two-stroke engine.
2. Perform baseline dynamometer tests on the standard 6V-71 to map its performance, emissions and fuel economy over full load and part load operating modes with diesel fuel.
3. Dynamometer test the 6V-71 operating with the prototype conversion kit and

optimize the conversion kit software and hardware for suitable performance emissions and fuel economy in both diesel and dual fuel modes.

1.3 OVERVIEW OF THIS INVESTIGATION

In this study, a Detroit Diesel 6V-71 transit bus two stroke engine was converted to dual fuel natural gas/diesel operation. The Electronic Dual Fuel Injection System of Alternative Fuel Systems Inc. was used as the basis for the conversion. High speed solenoid valves were employed to introduce natural gas into the cylinder through the air intake ports located on the circumference of the cylinder liners. The existing diesel fuel injectors were used to inject pilot diesel with the quantities controlled by an electronic rack actuator. The conversion kit was developed and optimized for this engine by preliminary testing. Following optimization, the engine was tested in both diesel only and dual fuel mode at maximum torque and over the 13-mode Japanese test. During the 13-mode test, the engine was run at steady state conditions of speed and torque for each of thirteen modes. All testing was carried out on an eddy-current type dynamometer in the Mechanical Engineering Department at the University of Alberta.

Chapter 2 describes pollutants from diesel engines and different methods of pollution control. Chapter 3 summarizes the characteristics of natural gas and its use as a vehicle fuel. It also deals with the performance and emissions of dual fuel engines and describes different dual fuel conversion systems based on a brief literature review of past studies done in the area. The dual fuel conversion system used in the study and the test equipment and procedure together with data reduction are described in Chapter 4. Tests results, analysis, and discussion are presented in Chapter 5. Chapter 6 gives the summary and conclusions obtained from this study, while raw experimental data and processed results are included in the appendices.

CHAPTER 2 DIESEL ENGINE EXHAUST EMISSIONS AND EMISSIONS CONTROL

2.1 DIESEL ENGINE EMISSIONS

In a diesel engine, fuel is sprayed under high pressure into air heated by virtue of its being compressed. The fuel burns and the piston is pushed downwards producing the work which is taken from the crankshaft. The diesel fuel consists of organic molecules which are mostly hydrocarbons. All hydrocarbons, on complete combustion in air yield carbon dioxide (CO_2) and water (H_2O). Unfortunately the heterogenous combustion occurring in a diesel engine is never perfectly complete. This leaves some quantity of producing partially oxidized products such as carbon monoxide (CO), hydrocarbons (HC), and particulate matter (PM). Diesel engines also emit nitrogen oxides (NO_x) generated in the combustion chamber by the reaction of nitrogen and oxygen from air. In practice, diesel fuel also contains some proportion of sulfur compounds, which burn to form sulfur dioxide (SO_2), together with traces of sulfur trioxide (SO_3).

Considering the number of possible reactions which could occur in a combustion chamber, the exhaust might contain hundreds of different compounds. For practical reasons a limited number of species are measured and regulated by legislation. Carbon monoxide, nitrogen oxides, non-methane hydrocarbons (NMHC), and particulates are the legislated diesel engine exhaust emissions.

2.1.1 Carbon Dioxide (CO_2)

Carbon dioxide emissions are a product of complete combustion of hydrocarbon fuel in air and are entirely controlled by the amount and the type of fuel burned. Assuming the same level of fuel consumption, fuel with a higher hydrogen to carbon ratio (H/C) produces smaller fractions of carbon dioxide. Although carbon dioxide poses no direct harm to human health, it is increasingly being identified as a "greenhouse gas" contributing to global warming. Carbon dioxide absorbs the reflected infrared energy from sunlight that strikes the Earth. The resultant effect is a warmer earth atmosphere system accompanied by substantial climatic changes in many parts of the world.

2.1.2 Carbon Monoxide (CO)

Carbon monoxide is a very toxic gas. It reacts with the hemoglobin molecule in red blood cells, forming carboxy-hemoglobin, which reduces the efficiency of this oxygen-carrying molecule [8, 44]. Excess exposure is directly life threatening while increased levels of carboxy-hemoglobin may cause high blood pressure, stroke, and heart attack.

Carbon monoxide emissions, which result from incomplete combustion and dissociation of the fuel, are controlled primarily by the fuel-air equivalence ratio. Carbon monoxide concentrations in the exhaust increase steadily with increasing equivalence ratio, as there is insufficient oxygen to convert all the carbon in the fuel to carbon dioxide. Since diesel engines mostly operate on the lean side of stoichiometric, they are expected to produce low carbon monoxide emissions. However, the lean CO emissions can be much higher than suggested due to locally rich mixture, or high temperature equilibrium and fast quenching at the low exhaust temperature of a diesel.

2.1.3 Oxides of Nitrogen (NO_x)

Oxides of nitrogen contribute to the production of ozone (O_3) in the lower atmosphere and to the acid precipitation, as well as they aggravate respiratory problems [8, 11].

The principal source of NO is the oxidation of molecular nitrogen during the combustion process in the vicinity of combustion regions at high temperatures, as described below by the extended Zeldovich mechanism [52].

$$O + N_2 \rightarrow NO + N \tag{2.1}$$

$$N + O_2 \rightarrow NO + O \tag{2.2}$$

$$N + OH \rightarrow NO + H \tag{2.3}$$

However, if the fuel contains nitrogen, the oxidation of the fuel bound nitrogen is an additional source of NO. The Zeldovich mechanism is exponentially dependent on the combustion temperature, highly dependent on oxygen atom availability, and the residence time which is defined as the amount of time in the combustion cycle spent at high temperature. Due to the strong temperature effect, NO_x formation is closely tied to high loads and efficiency improvements. The injection timing, which influences in-cylinder peak temperatures and the residence time, is a very important parameter affecting NO_x emissions.

While both nitric oxide (NO) and nitrogen dioxide (NO_2) are formed as final products, NO is the predominant species emitted. NO turns to NO_2 in atmosphere but NO is formed preferentially at high combustion temperatures. In diesels NO_2 can be 10 to 30 % of the total exhaust oxides of nitrogen emissions [52].

NO formed in the flame zone can be converted to NO_2 as follows:

$$NO + HO_2 \rightarrow NO_2 + OH \tag{2.4}$$

Subsequently, conversion of this NO_2 to NO occurs as follows:

$$NO_2 + O \rightarrow NO + O_2 \tag{2.5}$$

unless the NO_2 formed in the flame is quenched by mixing with cooler fluid. This explains higher NO_2/NO ratio occurring at light load in diesel engines [31].

2.1.4 Hydrocarbons (HC)

Hydrocarbon emissions are products of incomplete combustion of fuel and engine oil. The two major causes of hydrocarbons emissions in diesel engines under normal operating conditions are overleaning at low load and undermixing of fuel at high load. Overleaning takes place when fuel close to the spray boundary has mixed to leaner than the lower flammability limit during the ignition delay period. This mixture will not autoignite or sustain a fast reaction front and can only be oxidized by relatively slow thermal-oxidation reactions which will be incomplete at the low cylinder temperatures at part load. In contrast, undermixing is associated with excess fuel that enters the cylinder under overfuelling conditions. This usually happens at full load, where an increased level of unburned hydrocarbons results from inadequate oxygen in the fuel spray zone.

2.1.5 Particulate Matter (PM)

The presence of soot particles in the exhaust of diesel engine has been of concern to engine designers for many years. Soot particulates constitute a large fraction of the particulate matter emissions, which are regulated by legislation. Particulate emissions are defined by the Environmental Protection Agency (EPA) as any solid or liquid matter that collects on a binderless glass fiber filter [39]. Therefore, particulates include not only the solid carbon (soot), but also sulfate emissions arising from the sulfur in diesel fuel, and the soluble organic fraction (SOF) consisting of unburned and partially burned fuel (about 10-30 %) and lubricating oil (about 70-90 %) absorbed on the soot. The mass and composition of particulate matter produced by diesel engine depends on the engine design, the mode of engine operation, the properties of fuel burned, and the ambient conditions.

Diesel engines always form soot in the spray region. Most of the soot is then burnt as it mixes with hot air. However, under high load condition, poor air utilization during combustion leads to the formation of excessive amounts of soot which can not be burned up prior to exhaust. Consequently, the excessive soot in the exhaust constrains the fuel-air ratio at maximum engine power. The soot formation can be reduced by optimizing the mixture formation parameters such as injection pressure, air intake swirl, or diesel injector nozzle configuration. Turbocharging and air charge intercooling also have potential to reduce particulates at high loads by increasing air density and providing high air-fuel ratios.

Because the soot and soluble organic fraction result from incomplete heterogeneous combustion, the diesel engine emits much higher levels of particulate matter than the homogeneous combustion type gasoline engine.

2.2 EMISSIONS CONTROL

The stringent emissions standards imposed on heavy-duty diesel engines are a result of increased public awareness of air quality and the possible health hazard associated with diesel exhaust emissions. These standards require reduction of carbon monoxide, unburned hydrocarbons, oxides of nitrogen, and particulates.

The task to minimize emissions over the whole range of engine operating conditions is very difficult, especially as there are fundamental trade-offs between emissions. The trade-off between particulates and oxides of nitrogen emissions is a good example, in that combustion parameters which reduce one have the tendency to increase the other.

There are several strategies that may be employed to reduce emissions production including further refinements of the diesel fuel injection system, air flow modifications such as higher swirl, introduction of low sulfur diesel fuel, the use of the exhaust gas recirculation (EGR). Moreover, exhaust aftertreatment devices such as particulate traps and oxidation catalytic converters may be required in meeting the future emissions standards. The use of clean burning alternative fuels such as natural gas is also considered as a likely technical approach to meet the demanding emissions standards. The following sections describe briefly different approaches to lower emission levels from diesel engines.

2.2.1 Diesel Fuel Injection System

Diesel fuel injection system, being responsible for proper mixing of diesel fuel with the cylinder air, is the primary segment of the diesel engine emission control system. The main parameters of the injection system responsible for performance and exhaust emissions of the engine are the injection pressure, the injection timing, size of the nozzle spray holes, and the nozzle sac volume.

Injection pressure is an important parameter responsible for in-cylinder soot formation. Size of the diesel fuel droplets as well as the rate of fuel-air mixing, which largely controls the combustion rates during the diffusion burning portion of combustion, can be controlled by injection pressure. It was observed that carbon particulates decrease with increasing injection pressure [18, 39]. This is due to the fact that less fuel is burned in the diffusion combustion mode where mainly particulate formation occurs. However, increasing injection pressure must be done with considerable thought. Generally, as fuel injection pressure increases a more homogeneous, thoroughly mixed charge is produced and a penalty is paid in increased NO_x emissions.

Smaller spray holes improve fuel-air mixing which reduces soot emissions and keeps specific fuel consumption constant despite the longer duration of injection. Lengthened injection duration reduces NO_x formation and provides best atomization for soot control. Thus, a fuel injection system has to be designed for the smallest possible spray hole area in conjunction with the maximum useful injection pressure [29].

Injection timing is the most effective tool by which NO_x emissions can be controlled. Unfortunately, retarded injection timing is in conflict with good fuel economy and low particulate emissions. The use of an electronically controlled injection system to vary injection timing over the speed and load range would help to optimize fuel economy while meeting required emissions levels.

One important factor in formation of the fuel-derived SOF fraction is the nozzle sac volume [36]. As shown in Figure 2.1, the nozzle sac is the portion of the nozzle fuel passage between the nozzle needle and orifice. Fuel enclosed in the nozzle sac volume is exposed to combustion temperatures during each cycle and evaporates without an opportunity to be efficiently burned. Any reduction in the injector nozzle sac volume

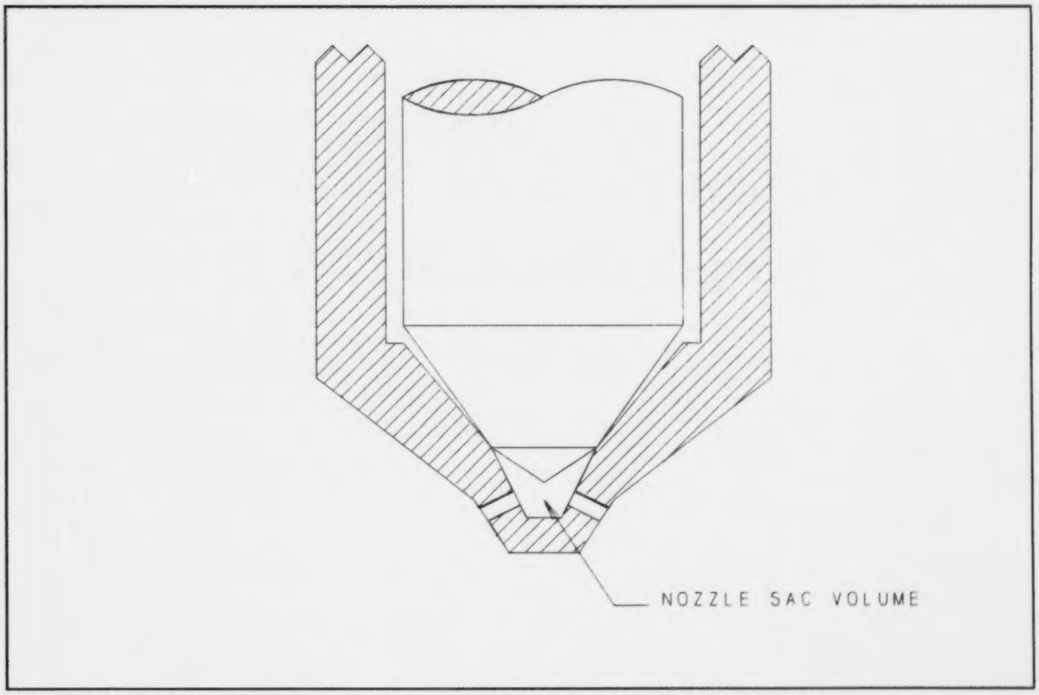

Figure 2.1 Nozzle sac volume in a typical diesel fuel injector.

directly improves SOF emissions.

2.2.2 Diesel Fuel

Diesel fuel quality has a significant influence on emissions. The fuel properties that have the largest impact are sulfur content, aromatic content, and cetane number.

The sulfur content in the diesel fuel contributes to pollution in two ways. First, during the combustion of diesel fuel over 95 % of the fuel sulfur is converted to sulfur dioxide (SO_2) and the remainder exits the engine as sulfur trioxide (SO_3), sulfuric acid (H_2SO_4), and sulfates. Since sulfates have the property to adsorb hydrocarbon components from the gas phase, the organic soluble particulates increase with increasing fuel sulfur content [18]. This means that the sulfur content of diesel fuel has a direct impact on the engine particulate emission level. To achieve the future particulate standards the sulfur content needs to be reduced to 0.05 % by weight. Until low sulfur diesel fuel is commercially available throughout Canada, the engine technology will not allow the 1994 U.S. standards to be achieved. Moreover, experimental results indicate that there is an inverse relationship between sulfate emissions and the performance of catalytic aftertreatment devices [28]. Thus, low sulfur fuel is the key element of the diesel engine emission control need for the 1990's.

Aromatics hydrocarbons constitute as much as 25 to 45 % volume of the No.2 diesel fuel and are known to contribute directly to the formation of particulates [41]. This is primarily due to the fact that the aromatic ring compounds are harder to oxidize. Test results indicate that aromatic content also increases NO_x emissions [35]. To achieve both low particulates and NO_x emissions a 10-20 % aromatic content would be desirable in the future diesel fuel.

Reduction of NO_x emissions can be also achieved by using diesel fuel with improved cetane number [41]. The NO_x reduction is generally attributed to less violent burning experienced with shorter ignition delay periods associated with high cetane number fuels. At light load conditions, lower particulate emissions are produced with high-cetane fuel due to more complete combustion and increased avoidance of lean flame-out conditions. At high load, however, larger quantities of diesel fuel are used and the shorter ignition delay period does not allow the diesel fuel to mix thoroughly with air by the time the combustion process starts. Consequently, increased levels of CO and soot emissions are produced.

2.2.3 Oxidation Catalytic Converter

It is not thought possible to meet the future emissions standards without some type of exhaust aftertreatment device. Diesel engines operate lean and have excess air in their exhaust. Therefore, an oxidation catalyst can be used to reduce tailpipe emissions on heavy-duty diesel engines.

Platinum or palladium are the most commonly used diesel oxidation catalysts. They promote the reactions of hydrocarbons (HC), liquid soluble organic fraction (SOF) and carbon monoxide (CO) with oxygen (O_2) available in diesel exhaust to form carbon dioxide (CO_2) and water (H_2O). The global oxidation reactions are presented by equations 2.5 and 2.6.

$$2CO + O_2 \rightarrow 2CO_2 \tag{2.5}$$

$$C_nH_m + (n + \frac{m}{4})O_2 \rightarrow nCO_2 + \frac{m}{2}H_2O \tag{2.6}$$

Diesel oxidation catalysts (DOC) reduce SOF as well as HC and CO. Depending on the type of engine and its exhaust, they oxidize 30-80 % of the gaseous HC and 40-90 % of the carbon monoxide (CO) present [37]. Diesel oxidation catalysts usually do not alter oxides of nitrogen emissions (NO_x) and have little effect on soot (carbon), but engine tests show that they typically remove 30-50 % of the total particulate load. This is achieved by oxidizing 50-80 % of the SOF present [37].

However, there are problems associated with the use of diesel oxidation catalysts. They convert sulfur dioxide (SO_2) to sulfur trioxide (SO_3), which reacts with water (H_2O) and forms sulfuric acid droplets (H_2SO_4) or solid sulfate particles. The creation of sulfur trioxide (SO_3) and sulfuric acid (H_2SO_4) is illustrated by equations 2.7 and 2.8 respectively.

$$2SO_2 + O_2 \xrightarrow{Catalyst} 2SO_3 \tag{2.7}$$

$$SO_3 + H_2O \rightarrow H_2SO_4 \tag{2.8}$$

The sulfate particles add to the amount of insoluble particulates emitted. Thus, using high sulfur diesel fuel in conjunction with oxidation catalysts leads to the increase of insoluble particulate mass emissions rather than to their reduction. Oxidation catalysts are also unable to control NO_x emissions in the lean diesel exhaust environment and they require further development.

2.2.4 Particulate Trap

Diesel particulate emissions are a major air quality concern, and one for which control is very difficult. The diesel particulate size ranges between 0.1μm and 10 μm, so particulates are easily dispersed in the ambient air and, if inhaled, can have a negative impact on human health [14]. Thus, it is necessary to reduce the particulate emissions of diesel engines down to the lowest technically available level. The particulate trap appears to be the leading candidate for particulate aftertreatment in heavy duty diesel engines.

A particulate trap generally uses a ceramic separation medium with tiny pores that capture particulates. The trapped mass is mostly dry soot plus the adsorbed SOF. As trapped material accumulates, flow resistance through the trap increases and smaller particles are removed until resistance is so high that regeneration is needed. Flow resistance equates to a pressure drop across the trap that increases engine back pressure and fuel consumption. When the resistance to flow is too high, the particulate trap has to be regenerated by heating it to burn off the soot. To sense when the trap is loaded with soot and to command the burner to start and stop the trap regeneration process, an auxiliary electronic control system is required. The external heat is necessary because the diesel exhaust temperature (150°-450°C) is significantly lower than the diesel soot ignition temperature (550°-650°C). The additional energy is usually supplied by an electrical heater or a diesel-fuelled burner placed in front of the particulate trap.

Trap efficiency to collect soot lies between 60 and 90 %. However, the regeneration process adds significant cost and complexity, and is the major obstacle to widespread use of particulate traps. Hence, new concepts in particulate trap technology are required to solve the regeneration problem.

2.2.5 Exhaust Gas Recirculation (EGR)

With the advent of stringent NO_x standards for heavy-duty diesel engines, exhaust gas recirculation (EGR) is being increasingly considered for NO_x reduction. The addition of diluents to the intake air reduces the peak flame temperature by increasing the specific heat capacity of the unburned zone, which consequently decreases NO_x emissions.

Results of many researchers showed that at very light load EGR is most effective in NO_x reduction and also conducive to reduced particulates [29]. The latter is due to both higher charge temperature and the fact that at low loads, particulates consist to a very high percentage (up to 90 %) of soluble organic fraction (SOF) and very little soot. Moreover, because of the high excess air ratio at low loads, smoke is generally very low and hardly affected even with high EGR rates.

However, with higher loads EGR reduces the overall air-fuel ratio which causes a sharp increase of particulates, especially the carbon soot fraction. Thus, a very accurate EGR metering system is required to control NO_x without excessive particulates generation at high load condition. The EGR rate has to be metered as a function of speed and load in order to meet the right balance between NO_x reduction and particulate increase. The fulfillment of this task is very difficult, especially during transient operation. Therefore, an EGR control system is often combined with an oxidation catalyst to improve the particulates-NO_x trade-off.

Another problem associated with the use of EGR is reduced durability due to the increased wear. The mechanism of abnormal wear is directly attributable to the recycling of soot particles and the formation of sulfuric acid (H_2SO_4) through the reaction of condensed water on the cylinder walls with the oxides of sulfur (SO_x) [29]. Additionally, EGR degrades the lubricant and results in a shorter oil drain period.

2.2.6 Alternative Fuels

Alternative fuels are widely regarded as a very likely approach to reduce emissions from heavy-duty diesel engines in the near future. Alcohols (methanol and ethanol) and compressed natural gas are considered as the most suitable alternative fuels for compression ignition engines [36, 37].

It was found that diesel engines powered by ethanol (C_2H_5OH) or methanol (CH_3OH) produce significantly lower particulate emissions than conventional diesels. This is mainly due to the relatively soot-free combustion of alcohol-air mixtures. Furthermore, alcohols do not contain any inorganic compounds, such as sulphur, thus sulfate particulates due to fuel are eliminated. Due to lower peak flame temperature of alcohols, caused by their high latent heat of vaporization (about five times that of diesel fuel), alcohol-fuelled engines emit fewer nitrogen oxides (NO_x) emissions. Many researchers observed that carbon monoxide (CO), gaseous hydrocarbons (HC), and aldehydes (RCOH) emissions are usually increased with alcohol operation [46, 47]. However, these emissions can be reduced to the desired level by using an oxidation catalytic converter.

Natural gas, being the cleanest hydrocarbon fuel available, is an even more promising alternative fuel for heavy-duty diesel engines. Because of the high hydrogen to carbon ratio (H/C), combustion of natural gas produces approximately 25 % less CO_2 than diesel fuel. This aspect is of great importance due to mounting concern with the "greenhouse effect". In addition, natural gas has a low propensity to soot and particulate production from engines using natural gas is negligible. Since natural gas contains a small amount of sulfur, sulfur dioxide emissions (SO_2) together with sulfate particulates can be significantly reduced. Nitrogen oxides emissions (NO_x), produced from diesel engines powered by natural gas, tend to be lower due to the low flame temperature of natural gas and burning under lean premixed conditions. In contrast, carbon monoxide and hydrocarbons emissions, consisted mostly of methane (CH_4), are higher than those from diesels.

In summary, natural gas and alcohols may be highly desirable vehicular fuels in markets driven by criteria such as emissions restrictions. The probability of meeting the future stringent nitrogen oxides (NO_x) and particulate emissions standards with a conventional diesel fuelled engine is uncertain.

CHAPTER 3 NATURAL GAS - ALTERNATIVE FUEL FOR COMPRESSION IGNITION ENGINES

3.1 PROPERTIES OF NATURAL GAS AND DIESEL FUEL

Natural gas, as the name implies, is produced in natural state from beneath the ground. It is a mixture of several gases, the main component of which is methane, which exists in concentration of 80 to 90 %. In fact, the composition of natural gas is not constant and can vary from source to source and also in the same site at different times. Non-methane hydrocarbons are composed mainly of ethane (C_2H_6) and in lower concentrations of propane (C_3H_8) and butane (C_4H_{10}). Concentrations of inert gases such as nitrogen (N_2) and carbon dioxide (CO_2) are the most common diluents present in natural gas. Table 3.1 shows the typical composition of natural gas coming from the University of Alberta main supply which was used during testing.

Table 3.1 Natural gas composition (University of Alberta - September 1993)

Natural Gas Composition	Volume Fraction (%)	Mass Fraction (%)
Methane	91.82	85.59
Ethane	2.91	5.08
Nitrogen	4.46	7.26
Carbon Dioxide	0.81	2.07
Total	100.00	100.00

Lower heating value = 45.27 MJ/kg Molecular weight = 17.21 kg/kmol

Properties of natural gas differ from those of conventional diesel fuel. Because natural gas is in a gaseous state at ambient conditions it must be compressed (CNG) to achieve a practical energy density for on-board storage. Typical compression pressures used at present are 16.5-24.8 MPa (2400-3600 psi). The volumetric energy content of natural gas stored at 3000 psi is 6.8 MJ/litre which is about 81 % less than the energy content of diesel fuel. However, the energy content of natural gas is about 5 % higher on a mass basis.

Diesel fuel contains chemically bonded sulfur, with the actual quantities depending upon the quality of the crude oil. As was mentioned in the previous chapter, sulfur is converted during combustion to sulfur dioxide (SO_2) which is environmentally incompatible due to its acidic reactions. In contrast, the sulfur compounds are not present in natural gas fuel because they are stripped from the gas near the point of production. Odorants added at distribution contain some sulfur, but its concentration is at ppm levels.

Natural gas is lighter than air and thus will dissipate harmlessly into the atmosphere if a leak should occur. The ignition temperature of natural gas is higher than that of diesel fuel and ignitable fuel-air ratio is within very narrow limits (5-15 % by volume in air). In addition, the minimum energy required to effect a successful ignition of natural gas-air mixture within the flammability region is relatively large (0.29 mJ). Natural gas, with an octane rating of about 120, has good anti-knock qualities and can be safely used with higher compression ratios.

Table 3.2 compares basic properties of the natural gas and Esso summer diesel fuel which were used for all tests.

Table 3.2 Properties of diesel and natural gas fuels.

Properties	Diesel Fuel	Natural Gas
Phase at ambient temperature	liquid	gas
Molecular weight	about 226	about 17.2
Composition	long chain hydrocarbons $C_nH_{1.8n}$	92% CH_4, 3% C_2H_6, 4% N_2, 1% CO_2
Specific gravity (@ 15 deg, 1 bar)	0.845	0.00073
H/C ratio	1.8	3.9
Stoichiometric A/F ratio (mass basis)	14.5-16.0	15.5
Flammability limits (% vol. in air)	0.6-6.5 %	5-15 %
Autoignition Temperature (@ 1 bar)	204-260°C	650°C
Lower Heating Value Mass (MJ/kg) Volume (MJ/l)	 43.17 36.5	 45.3 6.8[1]
Sulfur content (% mass)	0.24	N/A
Price[2] (cents/litre)	32.7	28.5 at the Fast Fill 11.6 for use by a VRA
Flame temperature (stoichiometric, STP)	2054°C	1875°C

[1] in storage conditions 15°C, 20.7 MPa.

[2] on energy equivalent comparison (Edmonton Transit prices from August, 1993)

3.2 ADVANTAGES OF NATURAL GAS AS C.I. ENGINE FUEL

Natural gas possesses several inherent advantages that make it attractive to use as the main supply of energy in highly efficient and reliable diesel engines rather than resorting entirely to refined liquid fuels. These include low pollutant emissions, availability, competitive price, combustion suitability, and safety features.

There are several clear environmental advantages to using natural gas in heavy duty diesel engines. In general, natural gas is considerable more benign than diesel fuel. It has potential to reduce carbon dioxide (CO_2), particulates, smoke, nitrogen oxides (NO_x), and sulfur oxides (SO_x) emissions. The reduction in NO_x and SO_x emissions reduces acid deposition by reducing the supply of acid precursors in the atmosphere. In addition, being compressed and stored in a "closed" fuel system, natural gas eliminates the evaporation problem of a liquid fuel system. If leak does occur, natural gas dissipates harmlessly into the atmosphere because it is lighter than air. Consequently, it does not contaminate soil and ground water, what would occur during a leakage of any liquid fuel. Compared with crude oil, natural gas production requires relatively little processing and can be utilized essentially the way it comes out of the ground. Hence, less pollution is involved in the production and refining of natural gas.

Natural gas resources are more plentiful than those of crude oil. Additionally, natural gas can be obtained from a variety of unconventional resources such as sewage and biomass, shale oil, and gasification of coal or peat. Thus, the use of natural gas as a vehicular fuel is an attractive way to decrease the dependence on imported oil and to assure a more secure energy future. As a consequence of vast resources and easier refining, natural gas costs less than other gaseous or liquid fuels in Canada. At the present time, 1 kg of natural gas costs 35.4 cents when purchased at a fast fill station, which corresponds to 28.5 cents per diesel equivalent litre[1]. This means that natural gas is about 13 % less expensive when compared with the actual price of 32.7 cents per litre of diesel fuel. However, when compressed by the user, the high compression cost fraction (59 % of the total cost at the fast fill station) is eliminated, and the cost of natural gas can be reduced to the level of 11.6 cents per diesel equivalent litre. Consequently, natural gas would be about 64.5 % less expensive than diesel fuel. This is especially attractive for large users, such as transit bus fleets, which could build their own natural gas fuelling stations and obtain large annual cost savings.

Efficient utilization of gaseous fuel in compression ignition engines depends primarily on its combustion properties. One of the most important factors is the fuel's resistance to knock. Methane, the main constituent of natural gas, has very good knock resistant qualities that make it an excellent alternative fuel for high compression ratio engine applications. However, a mixture of natural gas and air in the absence of a deliberate source of ignition will not ignite on its own, unless the intake air is preheated and an extremely high compression ratio is used [32]. Therefore, most of the practical applications of natural gas in compression ignition engines rely on an auxiliary source of ignition. Resorting to pilot diesel injection to provide ignition produces a very powerful and voluminous source of ignition; therefore, diesel pilot is well suited to igniting very lean mixtures of natural gas and air. The relatively slow flame propagation of natural gas is thus speeded up through the high compression ratio and the high ignition energy provided by the ignition of the pilot diesel. Hence, much leaner mixture operation is possible than in spark ignition engines. The gaseous nature of natural gas fuel contributes to a better engine transient operation by providing excellent mixing with the air, especially when gaseous fuel is introduced directly into the intake system. Moreover, the gaseous homogeneous nature of the charge will contribute to a smoother energy release by combustion following the ignition of the pilot diesel fuel. Since natural gas does not contain sulfur and does not mix with lubricating oil, it is expected that engines using natural gas will have lower maintenance cost and last longer than straight diesel engine.

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[1] Edmonton Transit price, August 1993

Natural gas has some excellent physical and chemical characteristics that make it clearly a safe automobile fuel. Because of its narrow flammability limit, natural gas will not propagate a flame in air at normal atmospheric temperature and pressure when the mixture concentration is less than 5.3 % or more than 15 %. Even when a leak occurs, unconfined natural gas quickly disperses to a concentration below its lower flammability limit. Moreover, a high ignition energy (0.29 mJ) and autoignition temperature (650 °C) make it difficult to start a fire with natural gas. As a vehicular fuel, natural gas is compressed and stored in cylinders constructed of steel, aluminium, fibreglass, or carbon fibre. Each cylinder is built to rigorous quality standards. It has to withstand pressure test, and some of them have to perform fatigue and fire immersion test. In addition, fuels lines are made of high strength steel. Consequently, the probability of accidental leakage as a result of collision is very small.

3.3 DISADVANTAGES OF NATURAL GAS AS C.I. ENGINE FUEL

Natural gas, comprised primarily of methane, has some disadvantages as a motor vehicle fuel. These include limited range, additional space and weight of natural gas cylinders, the cost of a conversion kit, and combustion limitations.

The main disadvantage of natural gas as a vehicle fuel, compared with diesel fuel, is reduced range. Consequently, the use of natural gas necessitates more frequent refuelling. This limitation in range is due to the low energy/volume ratio of the gas. One litre of compressed natural gas at 20.7 MPa contains about five times less energy than one litre of diesel fuel (6.8 MJ/l versus 36.5 MJ/l). This means that compressed natural gas (CNG) at 20.7 MPa occupies about five times the volume of diesel fuel on the energy basis. However, the limited range is not a serious problem for dual-fuel vehicles which can switch back to 100 % diesel operation when it is desirable to improve the travelling range.

The weight and bulk of the storage cylinders are often cited as drawbacks to natural gas use. The mass of tank depends mainly on the material of the gas cylinders, and can be significantly reduced by using modern composite materials. However, the mass and volume of natural gas cylinders does not appear to be as significant problem in an urban transit application because buses have space available for fuel tanks and the ability to refuel at a central location.

Further problems associated with the use of natural gas are the start-up barriers such as the cost of the conversion system and the cost of establishing new natural gas fuelling stations. At the moment, there is no large network of fuelling stations and the cost for a new CNG station is still high (about $250,000 per compressor plus $300,000 per station).

Dual fuel conversion of diesel engines can make possible the widespread substitution of natural gas for diesel fuel, with a significant reduction in emissions and fuel cost. Despite these advantages, two basic combustion problems are associated with conversion of diesel engines to dual fuel operation. First, with the high concentrations of natural gas, knock may occur as a consequence of spontaneous ignition at diesel compression ratios. This phenomenon limits the diesel replacement ratio. Second, if the concentration of the gaseous fuel is significantly below the lower limit of flammability, the gas may not react completely. This results in poor fuel utilization and in higher emissions of incomplete combustion products such as carbon monoxide (CO), and non-methane hydrocarbons (NMHC). In addition, a two-stroke diesel engine with homogeneous gaseous fuel-air mixture in the cylinder loses a portion of the gaseous fuel during the scavenging process. This reduces fuel economy and further raises emissions of incomplete combustion products.

3.4 STUDIES OF PERFORMANCE AND EMISSIONS OF DUAL FUEL ENGINES

Dual fuel engines of the compression ignition type represent in principle a very attractive method for the utilization of natural gas fuel for the production of power. They give relatively high efficiencies and outputs with relatively good exhaust emission characteristics. Several researchers have investigated the performance and emissions of dual fuel diesel/natural gas engine in the past. Based on their observations, this section describes combustion phenomena, performance and emissions at low and high load conditions, and knock phenomena of dual fuel engines.

3.4.1 Dual Fuel Engine - Principle and Combustion Phenomenon

The dual fuel natural gas/diesel engine is, mechanically, a conventional diesel engine of the compression ignition type. Some of the energy release comes from the combustion of natural gas while the diesel fuel pilot provides the remaining part of the energy release. The gaseous fuel is usually mixed uniformly with the intake air and compressed rapidly to below its autoignition conditions. Ignition of the charge is then initiated by injection of a small quantity of diesel fuel at some point near top dead centre. The natural gas/air mixture begins to burn by the action of multiple flame fronts swept through the mixture. The high energy provided by the pilot diesel is orders of magnitude greater than provided by a spark plug in a spark ignition engine [38]. This allows a dual fuel engine to operate at significantly leaner air-fuel ratios and at higher compression ratios than a spark ignition engine. Dual fuel engine output is usually controlled by adjusting both the concentration of natural gas in the air charge and diesel pilot. The premixed gas-air equivalence ratio is the key variable affecting both emissions and fuel economy. If the dual fuel engine uses the original diesel fuel injection system, it is also possible to retain the capability for full diesel fuel operation.

Combustion characteristics of dual fuel operation differ from those of straight diesel operation. In diesel operation the diffusion combustion is generally controlled by mixing. As a stream of diesel fuel is injected into the cylinder, it disintegrates rapidly into fine droplets which in turn vaporize and autoignite after mixing with high temperature compressed air. In dual fuel operation, the diesel pilot provides a great multitude of ignition sources for the premixed gas-air mixture. Once the mixture is ignited, the combustion tends to be mainly controlled by flame front propagation through the gas-air mixture.

The dual fuel engine can be either of the four-stroke or of the two-stroke type. However, a two-stroke engine is more demanding in its conversion to the dual fuel principle. The wasteful loss of the gaseous fuel to the exhaust with the air flow which scavenges the cylinder is the main problem associated with dual fuel two stroke engines.

3.4.2 Light Load Performance and Emissions

At light load, engine performance with dual fuel is usually inferior compared with diesel only operation. At light engine load, more total fuel energy for a given load is usually required with dual fuel operation. This is primarily a result of the increased ignition delay of the pilot diesel fuel with the addition of natural gas, which appears to inhibit the preignition process of the pilot fuel at very lean mixtures [23, 33, 40]. However, this ignition delay decreases with further gas addition thereby having less of an effect at higher engine loads [38]. As a result of lean air-gas mixture and the increased ignition delay at light load, a significant amount of natural gas can survive the combustion process unreacted. This is, in part, a result of the air-gas ratio approaching its lower flammability limit. Karim et al. [23, 24, 33] investigated this condition extensively. As the air-natural gas mixture approaches the lean flammability limit, the flame front initiated by the combustion of the diesel pilot no longer propagates through the entire combustion chamber, but becomes limited to the volume immediately surrounding the diesel pilot. A predominant fraction of the natural gas-air does not burn completely but undergoes pre-flame reactions until is exhausted from the cylinder. The major products of the pre-flame reactions are carbon monoxide (CO), and hydrocarbons (HC) [2, 33]. Consequently, dual fuel engines suffer from the presence of the unburned fuel in the exhaust, high hydrocarbons and carbon monoxide emissions, and poor fuel efficiency at light load conditions.

Similar results were obtained by Elliot and Davis who also studied the importance of gas-air mixture in dual fuel operation [48]. Their experiments with a CFR diesel engine showed that when the concentration of gas is below the lower limit of flammability the gas does not react completely with oxygen unless it is in, or immediately adjacent to an inflamed or high temperature region. The lower limit of flammability of natural gas in air under conditions existing at the end of compression was found to be approximately 4-5 % by volume.

The poor part-load performance of dual fuel engines can be improved by increasing the relative pilot diesel quantity, raising the intake air temperature, or throttling the intake air to provide a relatively richer mixture. Larger diesel pilot increases the overall charge temperature, provides more extensive ignition centres and a larger effective combustion zone. Advancing the injection timing of the pilot diesel enhances the dual fuel operation as well, by increasing the residence time and the activity of the partial oxidations reactions, and extending the flammability limit boundary [33]. As the intake air is partially restricted (throttled), less air is used to charge the cylinder and richer mixture is produced, for the same amount of gaseous fuel addition. The increased mixture strength leads to improvement of combustion and to reduction of emission of unburned hydrocarbons [43].

3.4.3 High Load Performance and Emissions

Many researchers observed that when relatively higher fuelling rates are employed the performance and emissions in dual fuel mode are significantly improved compared with those from low load operation [1, 40, 43].

Combustion of gaseous fuel can use the available air in the engine cylinder more efficiently than diesel combustion. One of the main limitations of the diesel engine is the fact that the power output is limited by excessive smoke produced from incomplete combustion near stoichiometric air-diesel fuel ratios. Although dual fuel engines show particulate emissions in the exhaust caused by diesel fuel combustion, the amount is reduced in comparison with diesel engines. By supplying part of the total fuel as a premixed lean mixture of natural gas with air, the fuel rich region formed by the diesel fuel spray is reduced, hence a reduction of the particulates. It has been shown that the premixed mixture combustion contributes very little to the soot emissions [42].

Barbour and others [20] have showed that the same four-stroke Oldsmobile 4.3 l V-6 diesel engine running in dual fuel mode using natural gas and diesel pilot produced 50 % fewer particulates under high load conditions. Even more promising results were obtained by Checkel et al. [1], who investigated performance and emissions of a Raba

2356 bus engine converted to dual fuel operation using an electronically controlled, multi-point, timed natural gas injection system. This engine emitted 91 % less soot over the Japanese six mode test while producing a 3.5 % more peak power in dual fuel operation.

Doughty, Bell, and Midkiff [40] evaluated performance and emissions of a four stroke Caterpillar 3406 turbocharged dual fuel engine. Their study showed that the efficiency of the dual fuel engine remains comparable to that of the pure diesel engine, when operating at high loads. The carbon monoxide (CO) emissions were higher while the carbon dioxide (CO_2) emissions were significantly reduced with natural gas dual fuelling. The sulfur dioxide (SO_2) emissions were reduced almost linearly with diesel fuel reduction. For full load dual fuel operation, about 15 % diesel pilot was used and the SO_2 concentration was about 15 % of the concentration obtained during straight diesel operation.

At high load, the equivalence ratio of the natural gas-air mixture used in dual fuel operation is much higher than the lean flammability limit; thus, there is no need to use a large diesel pilot. On the contrary, there appears to be a definite improvement in exhaust emissions associated with using a very small pilot. This means that higher diesel fuel replacement ratios can be achieved at high load conditions. Consequently, the pilot quantity can be reduced to 10 % or even less, depending on the engine parameters and type of the conversion system applied.

3.4.4 Knock Phenomenon in Dual Fuel Engines

One of the major problems of dual fuel engines is a tendency to knock during high load conditions. Since dual fuel engines use a preformed mixture of air and natural gas as a part of their energy supply, they are susceptible to knock. When very high power outputs or very high intake temperatures and pressures are involved, even a very knock resistant fuel such as natural gas will autoignite [34]. The incidence of knock poses an important limit on dual fuel engine power output and diesel fuel substitution ratio. Consequently, in some dual fuel engines, an increase of diesel pilot and reduction of gas concentration at high load conditions is required to avoid knock.

Knocking in dual fuel engines is due to autoignition of local regions of the natural gas-air mixture before the flame front arrival. The onset of knock is influenced by many factors including natural gas composition, effective compression ratio, combustion chamber design, natural gas-air mixture strength, intake temperature, injection timing and pilot diesel quantity [38]. Knock is usually associated with high rates of pressure rise, increase in heat transfer to the cylinder walls, and consequent loss in thermal efficiency [34]. Prolonged knock also leads to high component temperatures which can damage pistons, rings, and valves.

Karim et al. [24] investigated the influence of different parameters on knocking performance. An increase of the air intake temperature affects the onset of knock considerable. In fact, it has been shown that the knock limited power output is exponentially affected by the inverse of the absolute intake temperature. The onset of knock can be also delayed through lowering the induction pressures, retarding injection timing, and using leaner natural gas-air mixtures [23].

Wong and Messenger [12] converted the two-stroke Detroit Diesel 12V-149T turbocharged engine to dual fuel operation. At 1900 engine rpm and maximum power, severe engine knock occurred with 42 % of diesel fuel substitution. As a result, power output was lower by more than 15 % compared to diesel only performance. Analogous results were obtained by Lom and Ly [25] who modified one cylinder of the DDC 8V-92TA two-stroke diesel engine to run in dual fuel mode. Early in-cylinder injection of natural gas resulted in end gas knock. The addition of natural gas fuel was critical to autoignition for above 70 % of full load conditions. Sulatiski and Davies [22] tested John Deere's JC6466T four-stroke turbocharged dual fuel engine. A Mogas conversion system was applied which used a carburetor-mixer mounted in the engine intake system upstream of the turbocharger and intake manifold. The maximum diesel fuel substitution rate was limited by severe audible knock, so it ranged only between 42 and 48 %. Power reduction of 29 % occurred in switching from diesel to dual fuel operation as well.

3.5 OVERVIEW OF DUAL FUEL NATURAL GAS/DIESEL SYSTEMS

Metering the natural gas fuel and mixing it with the combustion air are very important functions in the dual fuel engine. Many approaches are possible, and the choice of approach has a major effect in determining the emissions and fuel efficiency of the dual fuel engine.

3.5.1 Single Point Fumigated System

The most common approach is to meter the natural gas into the intake air charge at a single point upstream from the intake manifold. The actual mixing of the gaseous fuel with air may be accomplished using a mechanical gaseous fuel mixer or an electronically controlled gaseous fuel metering system. Ideally, a homogeneous mixture of natural gas and air is created as the air-gas charge is delivered to the intake manifold.

The inherent deficiencies of single point fumigation systems are loss of power, poor fuel efficiency, high scavenging losses, low diesel fuel replacement, and susceptibility to knocking [22, 12]. Scavenge loss is an unavoidable feature of fumigated dual fuel systems due to diesel engine designs that result in air intake and exhaust valve open overlap at the start of the intake stroke. This scavenging process causes high natural gas losses in the exhaust, especially in two stroke dual fuel engines.

Sulatisky and Davis [22] equipped John Deere's JC 6466T four stroke turbocharged engine with a gas carburetion system developed by Mogas. The conversion used a special mixer mounted in the engine intake system upstream of the turbocharger and intake manifold. Because of severe audible engine knock, the diesel fuel substitution had to be kept below 48 %. In addition, power loss of 29 % was observed when the engine was switched from diesel only to dual fuel operation.

Wang and Messenger [12] investigated the feasibility of converting the two stroke Detroit Diesel 12V-149T turbocharged engine with compression ratio 17:1 to operate as a dual fuel engine using natural gas and diesel fuel. The natural gas fuel was injected into the air intake stream between the turbocharger and the intercooler. High unburned hydrocarbons and 15 % lower power outputs were encountered during dual fuel operation. The maximum power output and diesel fuel substitution ratio were limited mainly by engine knock, which occurred with diesel substitution ratios higher than 42 %.

Scavenging losses of natural gas as high as 38 % were measured during dual fuel operation at 1900 rpm engine speed. Additionally, an "after burn" phenomenon was encountered when richer mixtures were used and the unburned gaseous fuel from the scavenging process ignited within the exhaust system.

3.5.2 Multi-Point Injection System

In a multi-point injection system, the gaseous fuel is delivered separately to each cylinder just after the exhaust valves are closed, which reduces the possibility of scavenging losses. This configuration offers the advantage of making adjustments in air-fuel ratio very quickly to obtain low exhaust emissions and good fuel economy.

This approach is much easier to use in four stroke engines in which there is usually plenty of time for natural gas injection during the intake stroke after the closure of the exhaust valves. In contrast, in most two stroke engines there is little time for natural gas injection through the intake ports in the cylinder liner. Moreover, two stroke engines have high scavenging efficiency and the exhaust valves typically close after the intake ports are covered by the piston. Thus, an accurate metering and timing of the natural gas injection is critical in two stroke dual engines. The multi-point injection system is therefore considered as a suitable choice for two stroke engines because it offers the possibility of accurate natural gas metering and timing to reduce scavenging losses and pollutant emissions.

Checkel and co-workers [1] used an electronically controlled sequential multi-point natural gas injection system on the four stroke Raba 2356 bus engine. The system has achieved an excellent diesel replacement: 90 % at maximum torque, 75 % on the Japanese 6 mode test. Compared with the base diesel engine, a 3.5 % power increase was available in dual fuel operation. With the diesel injection timing left at the stock value, emissions of CO_2, CO and NMHC were reduced substantially, (28 %-41 %), but NO_x emissions were increased, (by 83 %). However, retarding the diesel pilot injection to the "minimum best torque" position gave a 30 % NO_x reduction.

3.5.3 Direct Injection System

An attractive approach for the conversion of diesel engines to dual fuel operation is the direct injection of natural gas as a gas jet into the cylinder after valve closure and well into the compression stroke. A small amount of diesel fuel is injected to ignite the natural gas.

The main advantages of the high pressure injection of natural gas directly to the cylinder are: no knock limit if gas injection is after or simultaneous with liquid fuel pilot injection, diesel cycle efficiency at all loads and speeds, and negligible unburned fuel in the exhaust [26]. However, there are also some disadvantages associated with the direct injection system. It requires a high pressure gas supply, (about 20 MPa), and relatively sophisticated and expensive gas injectors with electronic control. During diesel operation, the gas injector remains idle while being exposed to high temperature combustion gases without the benefit of injected gas cooling. In addition, many modifications to the cylinder head are usually required to accommodate a second injector. This approach requires further research and development efforts especially since a lack of uniform mixing of the injected fuel with the air, within the short time available, may lead to patchy "flame propagation" and increased exhaust emissions [33].

The work of Einang and co-workers [49] involved the direct injection of natural gas into a large bore (300 mm) two stroke single cylinder engine. Pilot fuel was injected by an unmodified diesel injector and gas was injected at 15 MPa through a separate injection valve fitted into the cylinder head. With 73 % (energy proportion) natural gas fuelling, thermal efficiencies were similar to 100 % diesel operation. In preliminary testing the natural gas injection gave slightly higher smoke readings than with only diesel fuelling and NO_x emissions were reduced 24 %.

Hodgins, Gunavan, and Hill [26] developed an intensifier-injector system for natural gas fuelling of a Detroit Diesel two stroke diesel engine. The concept employed direct injection of high pressure natural gas with pilot diesel ignition and an engine-driven intensifier compressor to pressurize the natural gas to the injection pressure of 20 MPa. The diesel pilot fuel and the high pressure natural gas mix as they enter the combustion chamber through a common nozzle. NO_x emissions with the intensifier-injection were generally reduced compared to 100 % diesel operation, while CO and HC emissions were increased resulting from incomplete burning due excessive dispersion of the gas during the pilot fuel ignition delay time. Diesel baseline thermal efficiency was exceeded at higher loads, while poor efficiency was observed at low loads.

3.6 PRESENT WORK VERSUS PREVIOUS STUDY

Natural gas has been successfully used in spark ignition engines, but its use in diesel engines has been limited primarily to low speed stationary engines. Most the diesel engines converted to dual fuel operation have been four stroke type, and the use of natural gas in the two stroke diesel engines was always considered more difficult. This is due to the fact that two stroke engine is much more demanding in its conversion, mainly because of fuel loss directly to the exhaust during the scavenging process. To date, very few tests have been reported over the full range of speeds and loads and very few complete emissions measurements have been reported on two stroke dual fuel engines.

Large users in the transport industry and bus operators have recently become aware of the potential benefits of using natural gas. Since a large number of heavy-duty trucks and buses are powered by two stroke diesel engines, there is a need of further research directed towards the satisfactory employment of two stroke engines for dual fuel engine application.

This work contributes to the research effort by developing a prototype dual fuel natural gas/diesel conversion system for the Detroit Diesel 6V-71 two stroke engine, and conducting complete emissions and performance tests.

CHAPTER 4 ENGINE, NATURAL GAS CONVERSION SYSTEM, TESTING EQUIPMENT AND PROCEDURE.

In this study, a two stroke Detroit Diesel Model 6V-71 transit bus engine was converted to dual fuel natural gas/diesel operation. Description of the two stroke cycle principle and specifications of the DDC 6V-71 engine are presented in Section 4.1. Section 4.2 describes the Electronic Dual Fuel Injection System used to convert the engine to dual fuel operation while the testing equipment is described in Section 4.3. The engine was tested (before and after conversion) at maximum torque over its speed range and the Japanese 13-mode test was used to evaluate part-load performance. Both testing procedures are described in Section 4.4 while data reduction is discussed in Section 4.5.

4.1 TWO STROKE CYCLE PRINCIPLE AND ENGINE DESCRIPTION

The test engine was a naturally aspirated uniflow two stroke Detroit Diesel Model 6V-71 transit bus engine. The engine is typical of those used in most urban buses in Canada.

In the two cycle engine, intake and exhaust take place during part of the compression and power strokes respectively. A blower is provided to force air into the cylinder to expel the exhaust gases and supply the cylinders with fresh air for combustion. The DDC 6V-71 engine uses intake ports in the cylinder wall and cam-driven exhaust valves in the cylinder head (see Fig. 4.3). The cylinder wall intake ports are above the piston when it is at the bottom of its stroke. These ports admit the air from the blower into the cylinder as soon as the rim of the piston uncovers the ports. The unidirectional flow of air towards the exhaust valves produces a scavenging effect, leaving the cylinders full of relatively pure air when the piston again covers the inlet ports. As the piston continues on the upward stroke, the exhaust valves close and the charge of fresh air is compressed. Shortly before the piston reaches its highest position, the required amount of diesel fuel is sprayed into the combustion chamber by the unit fuel injector. The diesel fuel ignites and the combustion continues until the whole fuel injected has been burned. The resulting pressure forces the piston downward on its power stroke. The exhaust valves open during the downward stroke, allowing the burned gases to blow down into the exhaust manifold. The downward moving piston then uncovers the inlet ports and the cylinder is again swept clean with scavenging air. This entire combustion cycle is completed in each cylinder for each revolution of the crankshaft, or, in other words in two strokes. Figure 4.1 shows the exhaust valve and air intake port timing diagram for the 6V-71 engine.

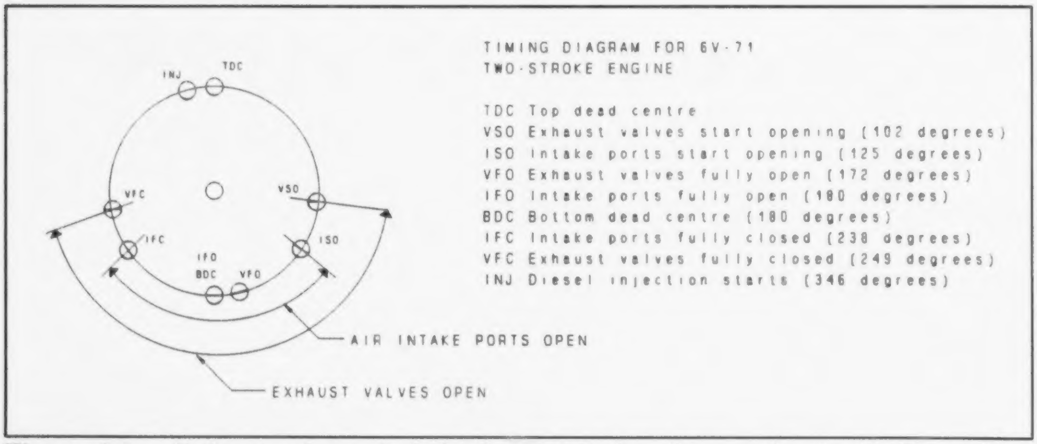

TIMING DIAGRAM FOR 6V-71
TWO-STROKE ENGINE
TDC Top dead centre
VSO Exhaust valves start opening (102 degrees)
ISO Intake ports start opening (125 degrees)
VFO Exhaust valves fully open (172 degrees)
IFO Intake ports fully open (180 degrees)
BDC Bottom dead centre (180 degrees)
IFC Intake ports fully closed (238 degrees)
VFC Exhaust valves fully closed (249 degrees)
INJ Diesel injection starts (346 degrees)
INJ
TDC
VFC
VSO
IFC
ISO
IFO
BDC
VFO
AIR INTAKE PORTS OPEN
EXHAUST VALVES OPEN

Figure 4.1 Exhaust valve and air intake port timing diagram for the 6V-71 engine.

The test engine was equipped with mechanical unit injectors with constant start of fuel injection and variable injection duration. A double weight mechanical governor was originally used on the 6V-71 engine to control idle speed and limit the maximum operating speed. Basic engine specifications are presented in Table 4.1.

Table 4.1 Detroit Diesel Model 6V-71 engine specifications.

Basic Engine	6V-71
Model	7067-4121
Engine Number	6VA-84625
Type	Two cycle, 63.5°V
Number of cylinders	6
Bore	108 mm
Stroke	127 mm
Displacement	6.9 litre
Aspiration	Naturally Aspirated, Roots Type Blower, No Turbocharger
Compression ratio	18.7
Exhaust valves per cylinder	2
Idle speed	570 rev/min
Maximum speed	2300 rev/min
Peak power	134 kW @ 2200 rpm
Peak torque	735 N.m @ 1000 rpm
Crankshaft rotation	Counter clockwise

A Detroit Diesel two-stroke engine equipped with a blower but without a turbocharger is called naturally aspirated engine. Even though the blower forces air into the cylinder, the exhaust valves are open after the intake ports close, so the pressure at the start of the compression stroke is set by the sum of the atmospheric and the exhaust back pressures. The nominal compression ratio of 18.7:1, defined by the manufacturer, is based on swept volume, bowl volume, and clearance volume. However, the effective compression ratio would be 15.4:1, assuming that compression starts after the intake ports are closed, or 14:1, assuming that compression starts after the exhaust valves are closed.

4.2 CONVERSION SYSTEM DESCRIPTION

The Electronic Dual Fuel Injection System developed with Alternative Fuel Systems Inc. was used to convert the 6V-71 engine for dual fuel diesel/natural gas operation. The conversion philosophy was to make a bolt-on kit rather than requiring extensive parts replacement and machining. Therefore, the existing compression ratio, blower drive, and diesel injection system were retained as much as possible. The conversion system consisted of gas injection hardware, diesel control hardware, and engine management software, all of which had to be developed or adapted for this engine. The system enabled the stock engine to operate on natural gas using diesel pilot to ignite mixtures of natural gas and air. To obtain maximum diesel fuel displacement and low pollutant emissions, optimum control of both fuels over all speed and load conditions was necessary.

The key element is a high speed solenoid valve (HSV) which performs the function of metering, timing, distributing, and injecting the gaseous fuel to each cylinder. The valves were supplied with natural gas at 2068 kPa (300 psi) and activated near the end of the scavenging process when the air intake ports were still open. To control the diesel fuel supply, the mechanical governor was disconnected and its function was taken over by an electronically controlled rack actuator, which was installed on the top of the governor housing. No changes were made to the cylinder head or engine block and the original diesel injectors were used during both diesel only and dual fuel operation. Standard injection timing was used throughout all tests. The gas injection system and rack actuator acted together to control dual fuel operation. The rack was positioned to provide exclusively diesel operation for idle and low power operation. At moderate and high power levels, the rack was positioned to provide a reduced diesel quantity and the solenoid valves were activated to supply a large part of the energy requirement with natural gas.

An important feature of the conversion system is that no major engine modifications are required and normal engine performance (torque and power) is maintained in both dual fuel and diesel only operation. As well as being easy to install, the conversion system allows the operator to bring the engine back to 100 % diesel operation any time it is necessary. The basic components of the AFS Electronic Dual Fuel Injection System are:

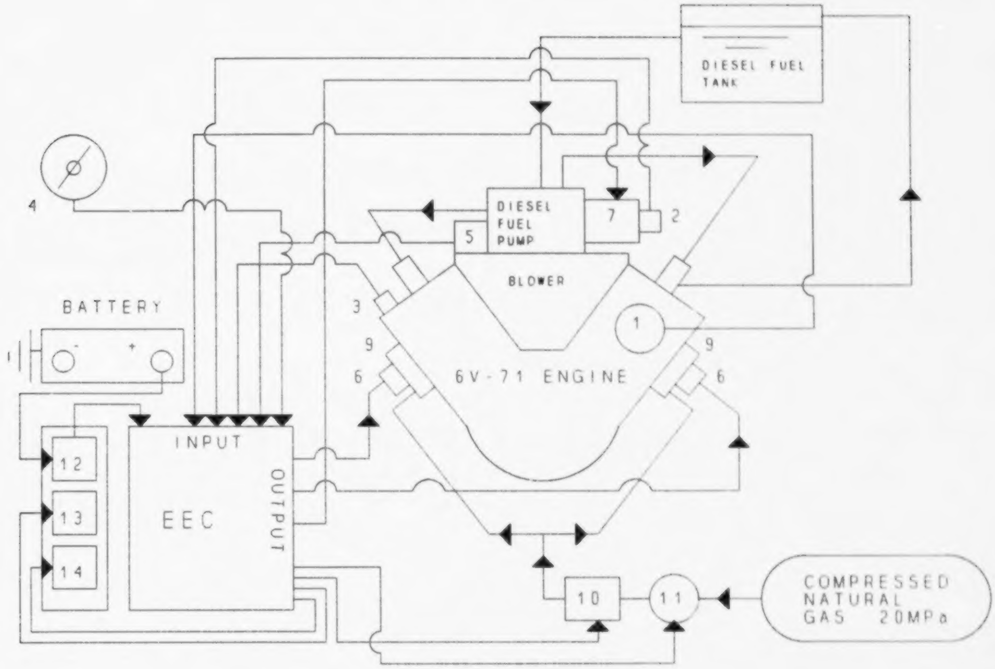
- a microprocessor, an electronic control unit
- a set of HSV gas injection solenoid valves, one per cylinder
- the electronically controlled rack actuator ("ractuator")
- two gas injection manifolds which position the solenoid valves
- a Hall effect timing sensor

- air and coolant temperature sensors
- manifold absolute pressure sensor
- "throttle" position sensor for engine control.

The conversion system operates on 12 volts from the engine power supply. Figure 4.2 shows a schematic of the Electronic Dual Fuel Injection System as applied to the 6V-71 engine.

The Hall effect timing sensor was installed on the top of the camshaft pulley, (which turns at the same speed as the engine crankshaft). The Hall effect sensor's mask provides 12 input signals per cycle for speed measurement and one input signal per cycle for position measurement. It senses a magnet when the # 1 cylinder is at top dead centre. An automotive electronic "throttle" position sensor is used to sense desired engine load. The high speed solenoid valves were installed in two special manifold blocks mounted on the sides of the engine block. Each manifold was bolted to the engine block, in the place of the inspection covers. A short tube was used to deliver natural gas from the solenoid valves in the manifold block directly to one cylinder liner intake port on each cylinder. Two different sets of tubes were produced. The first set of tubes directed the natural gas upwards as it entered the intake port while the second set directed the gas down towards the piston. Figure 4.3 shows a schematic of the engine with the upwards pointing tubes installed.

There is another modification roughly equivalent to a crankcase ventilation system. The air box drain tubes, which normally vent to atmosphere, were connected to the oil trap container and then rerouted back to the engine intake manifold, ahead of the engine


DIESEL FUEL TANK
4
DIESEL FUEL PUMP
5
7
2
BLOWER
BATTERY
3
9
6
1
9
6
6V-71 ENGINE
INPUT
12
13
14
EEC
OUTPUT
10
11
COMPRESSED NATURAL GAS 20MPa

AFS DUAL FUEL SYSTEM

1. Hall Effect Sensor
2. Rack Position Sensor
3. Engine Coolant Temperature Sensor
4. Throttle Position Sensor
5. Air Charge Temperature Sensor
6. Intake Absolute Pressure Sensor
7. Rack Actuator
8. High Speed Valves
9. Gas Manifold
10. Shut-off Solenoid Valve
11. Gas Pressure Regulator
12. Power on/off Switch
13. Dual Fuel Indicator Light
14. Mode Select Switch

Figure 4.2 Schematic of the AFS Electronic Dual Fuel Injection System on the 6V-71 Engine.

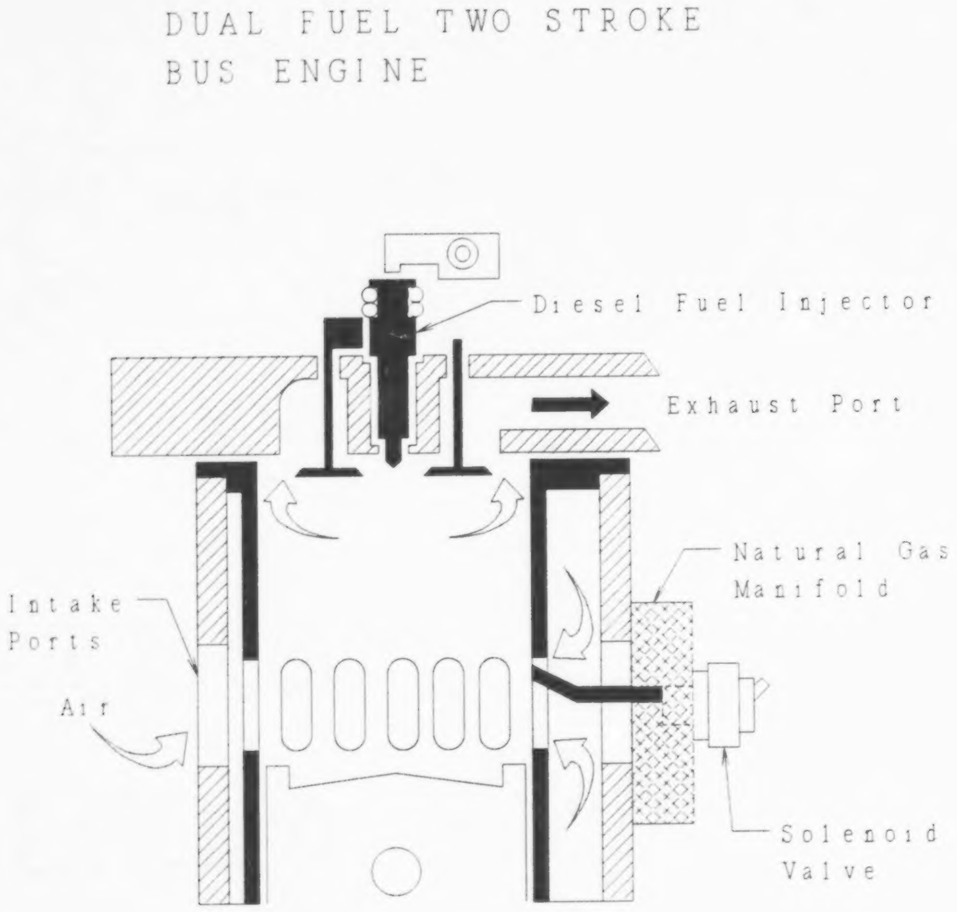

DUAL FUEL TWO STROKE
BUS ENGINE
Diesel Fuel Injector
Exhaust Port
Natural Gas
Manifold
Intake
Ports
Air
Solenoid
Valve

Figure 4.3 Schematic of the converted engine (with the upwards pointing tubes).

blower. This was necessary to avoid the possibility of natural gas leakage from the engine air box to the testing room.

4.2.1 Description and Calibration of High Speed Natural Gas Valves

The gaseous fuel was delivered to each cylinder by individual high speed solenoid valves which controlled the timing and volume of injected natural gas. The valves used were Servojet Model HSV 3000, which specifications are presented in Table 4.2.

Table 4.2 Servojet Model HSV 3000 high speed solenoid valve specifications.

Producer	Servojet Product International
Type	3 Way Normally Closed
Material	Stainless Steel
Pressure Rating	2068 kPa (300 psi)
Life Expectancy	2 billion cycles
Temperature Range	-40 to +149 °C (-40 to +300 °F)
Speed	up to 200 Hz
Response Time	2 milliseconds
Timing Precision	± 25 microseconds

Since gas flow is sonic across the valve inlet seat when opened, mass flow rate is approximately proportional to inlet pressure. In this study, the solenoid valves were supplied with natural gas at 2070 kPa (300 psi) and the pressure was kept constant by a pressure regulator. Any drifts in the supply pressure could decrease or increase fuel supply to the engine.

The amount of natural gas injected into the cylinder was controlled electronically by the solenoid pulse width. Thus, a calibration determining the relationship between solenoid energize time (pulse width) and the quantity of natural gas that enters the cylinder was necessary for proper operation of the software controlling the dual fuel operation. The high speed solenoid valves were calibrated at the supply pressure of 2070 kPa (300 psi) on the engine running in dual fuel mode. The natural gas mass flow rate and engine speed were measured to calculate natural gas flow in milligrams per injection as a function of pulse width in milliseconds. In the case of the 6V-71 system, the engine was run at a constant speed of 1000 rpm while the pulse width was varied. The resulting calibration flow chart can be found in Figure 4.4. During dual fuel operation, the quantity of natural gas (in milligrams per injection) calculated by software is converted to gas injector pulse width (in milliseconds) by interpolation between the two closest calibration points. The average correlation coefficient between the gas injector pulse width and the natural gas quantity (based on four calibration points) was found to be 4.867 mg/(inj × ms).

4.2.2 Rack Actuator Calibration

A linear actuator ("ractuator") which is controlled by the microprocessor was used to position the diesel fuel rack in place of the conventional mechanical governor. It is responsible for the metering of diesel fuel in both dual fuel and diesel only operating modes. A calibration of the rack actuator was necessary to determine the relationship between the quantity of fuel injected and the position feedback signal for all engine speeds. Once this relationship is known, the computer can accurately position the rack

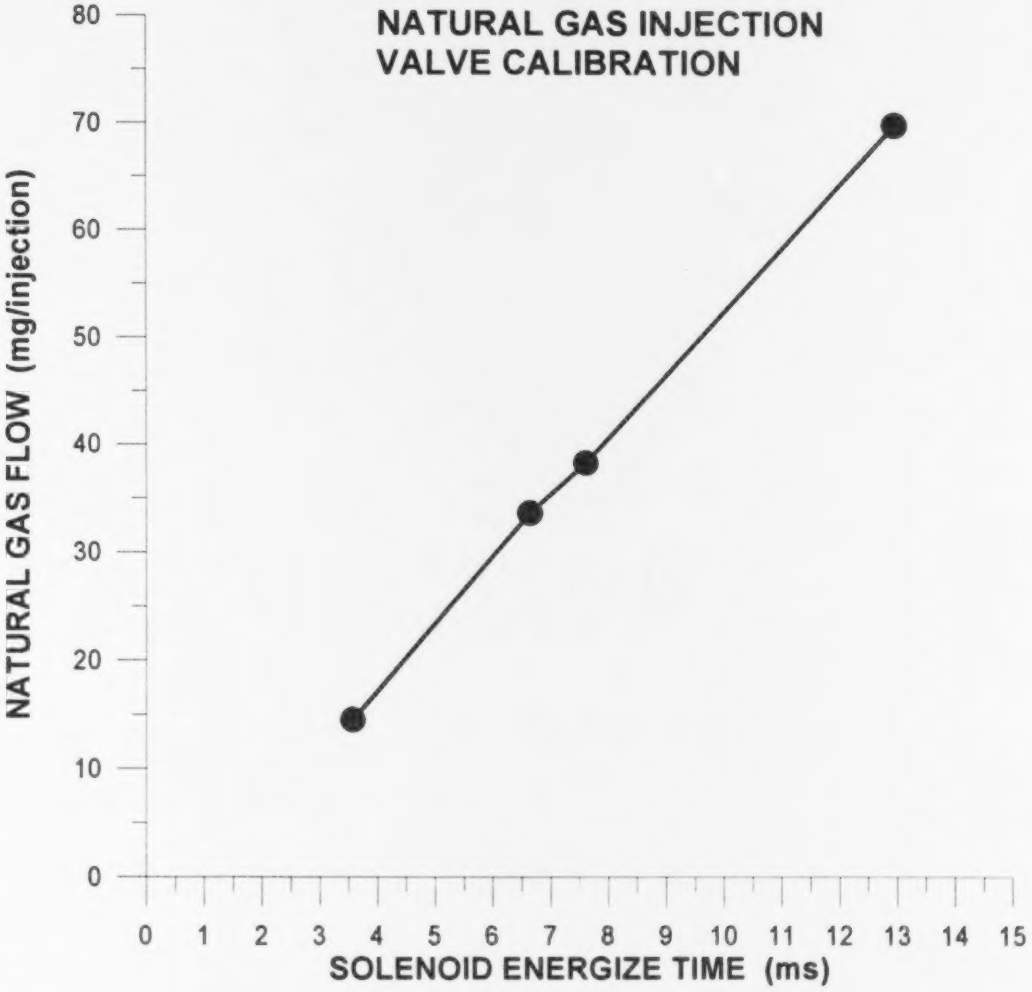


Figure 4.4 Natural gas injection valve calibration.

to obtain any desired quantity of diesel fuel. A potentiometer was used for the rack position feedback.

In the case of the 6V-71 system the rack actuator was calibrated for the whole range of load at the following engine speeds: 550, 600, 800, 1000, 1400, 1800, 2200, and 2300 rpm. During the calibration the following parameters were measured: engine speed, diesel fuel mass flow, diesel fuel temperature, and linear travel of the rack actuator. Based on these measurements, the relationship between the rack actuator position and the quantity of diesel fuel in cubic millimetres per injection (mm^3/inj) was established for each calibration speed. The resulting calibration curves can be found in Figure 4.5 and the calibration data were tabulated and included in Appendix A.

The calibration showed that accurate and repeatable control of the diesel fuel flow rate was achieved by the use of the electronically controlled rack actuator. Complete control over the diesel fuel delivery from zero to maximum was accomplished, and no internal modifications to the diesel fuel injection system were required.

4.2.3 Engine Mapping

For optimum engine performance and low exhaust emissions, it is necessary to vary the quantity of diesel pilot and natural gas over all operating conditions. During the optimization of the control system, a wide range of operating points were determined by the process of engine mapping. Engine mapping involved operating the engine at a fixed speed and load and varying control variables to determine their optimum values. In the case of the 6V-71 system, the control variables were diesel fuel quantity (diesel pilot), natural gas fuel quantity, and natural gas injection timing.

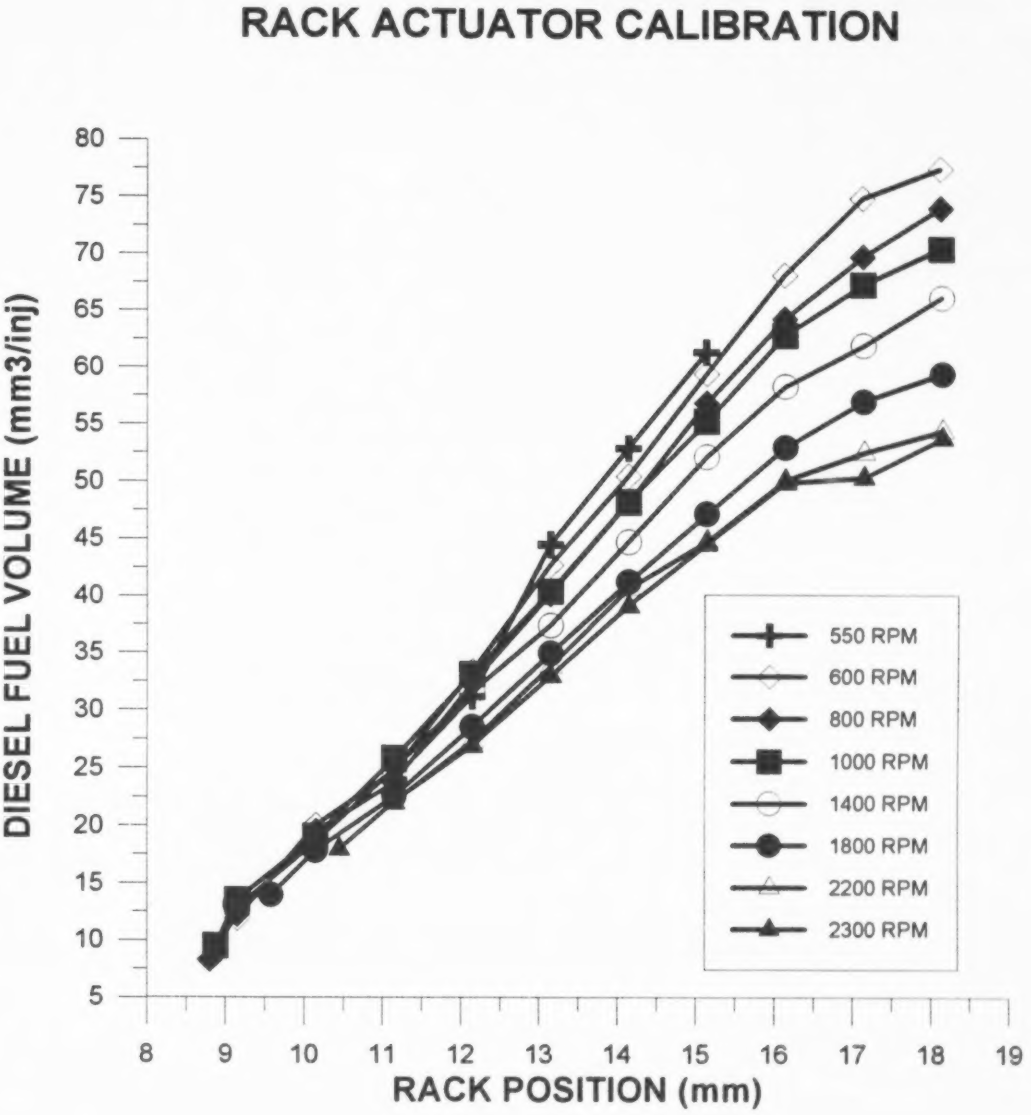


Figure 4.5 Rack actuator calibration.

The diesel pilot was set to optimize engine efficiency, diesel substitution, and exhaust emissions and at the same time ensure that the engine did not operate in a regime where durability would be reduced. It was found that diesel pilot quantity had a significant effect on hydrocarbon emissions. When the diesel pilot quantity fell below a threshold level of 11 mm^3 per injection, combustion of the natural gas failed to occur and high hydrocarbon emissions were observed. Consequently, at idle and low loads when total commanded fuel was less than 11 mm^3 per injection, the engine was set to operate on straight diesel. To prevent misfire due to an overly lean mixture of natural gas and air at light load conditions, a significant fraction of the total fuel had to be delivered by diesel pilot. In contrast, more natural gas could be used at moderate to high loads and lower engine speeds. The influence of engine speeds was mainly due to high intake air charge temperature coming from the blower. The intake air charge temperature has a strong influence on engine knock, and knock limits the maximum natural gas usage. Figure 4.6 shows that the air intake charge temperature rises dramatically at higher engine speeds.

The optimum diesel pilot values for all operating point conditions tested were tabulated in a pilot map which was saved in the software that controls dual fuel operation. The diesel pilot map is included in Appendix A. Figure 4.7 shows a graphical representation the diesel pilot map. The figure gives the ratio of total natural gas (trapped and scavenged) to the total commanded fuel and engine speed during dual fuel operation.

In this study, the natural gas fuel was injected to the cylinder through the intake

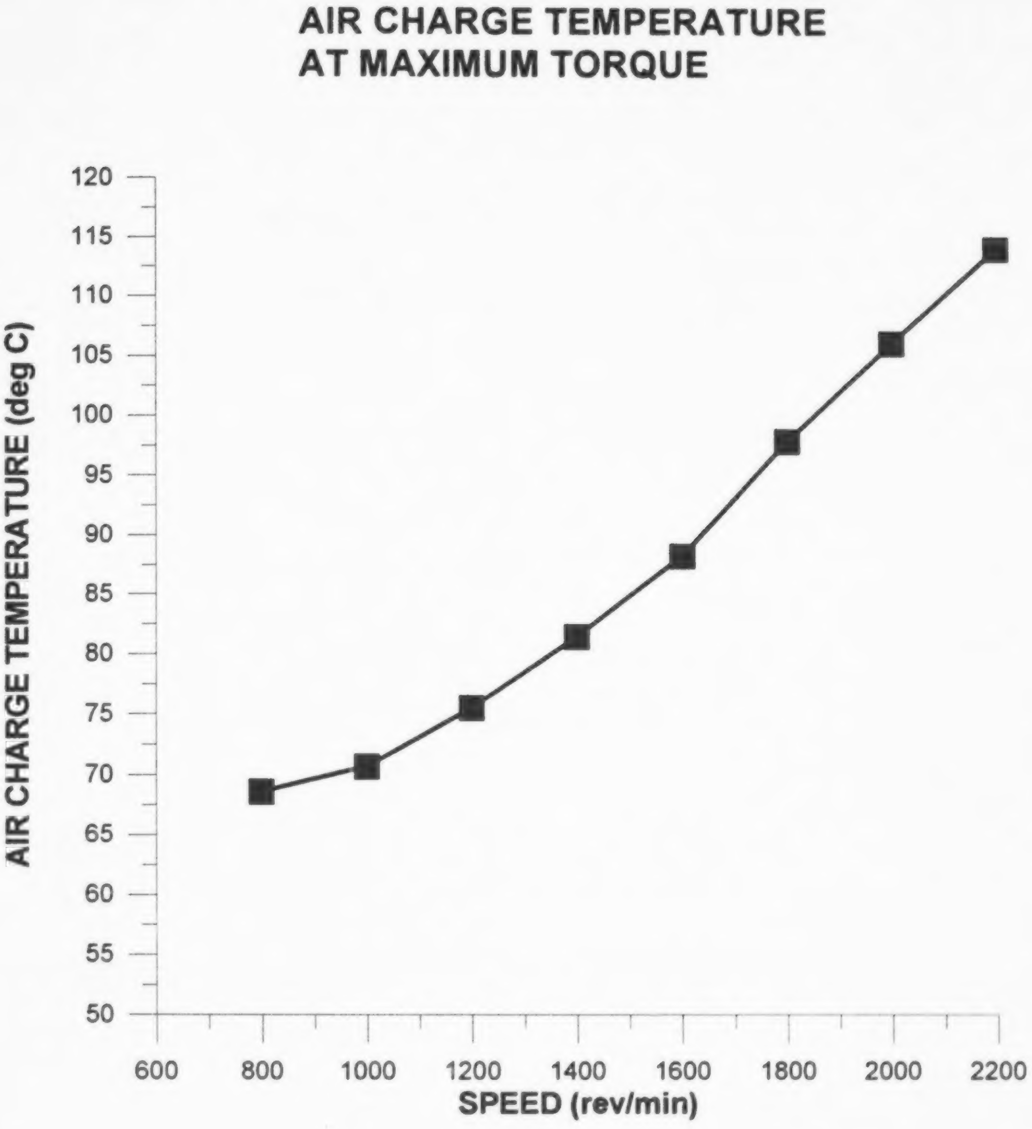


Figure 4.6 The air charge temperature as a function of engine speed at maximum torque.

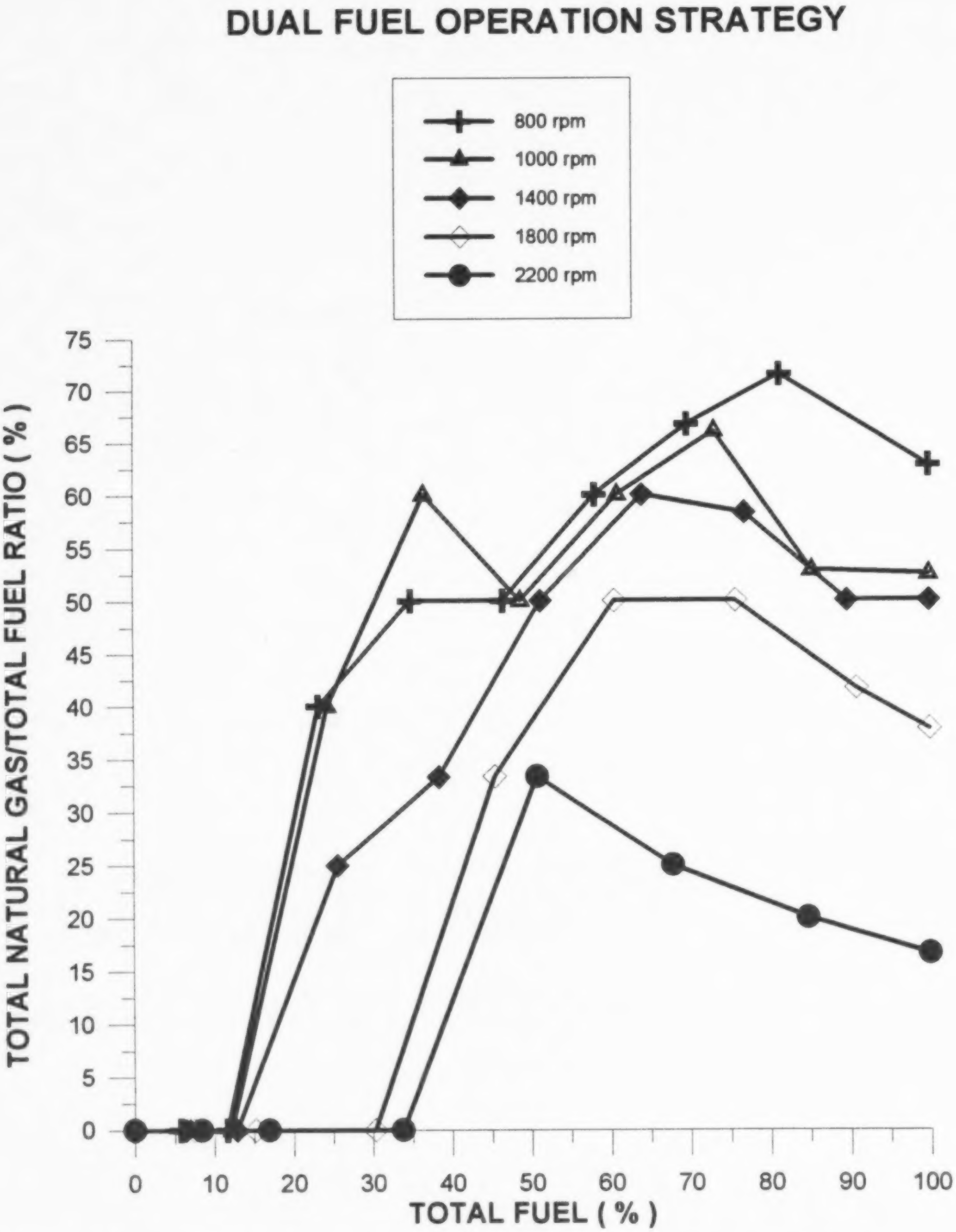


Figure 4.7 The ratio of total natural gas (trapped and scavenged) to total commanded fuel as a function of total commanded fuel and engine speed.

ports in the cylinder liner. Therefore, injection is restricted to the period during which the intake ports are uncovered by the piston. However, as was presented in Figure 4.1, for correct scavenging in the two stroke engine, the exhaust valves start opening while the air intake ports are still closed and the exhaust valves remain open after the intake ports are already closed. This means that the exhaust valves are still open when natural gas is injected into the cylinder, through the intake ports. In order to minimize the loss of gaseous fuel due to the scavenging process, timed injection of natural gas is necessary. It is essential to inject the gas as late as possible before the intake ports close so that it is neither blown through the engine with the scavenge air nor trapped outside the cylinder by the closing port. Therefore, extensive testing was conducted to develop a gas valve timing map over the full range of engine operating conditions. During this testing, natural gas injection timing was set by both the tendency of gas to be swept into the exhaust system by the scavenge air and the changes in engine torque output. Generally, optimum torque output directly correlated to minimum hydrocarbon emission rates. Figure 4.8 shows an example of how torque output varies with natural gas injection timing at constant engine speed and constant natural gas flow. As a result of this testing, the natural gas timing map, containing a variable end-of-injection point (in terms of crank angle degrees) for each given speed, load, and gas delivery tube configuration, was produced. Ideally, the end-of-injection point would determine that the gas finished leaving the delivery tube just as the piston was about to cover the intake ports in the cylinder liner. However, to keep the centre point of the gas injection timing at the optimum injection point determined by the lowest gas losses, the end-of-injection was set

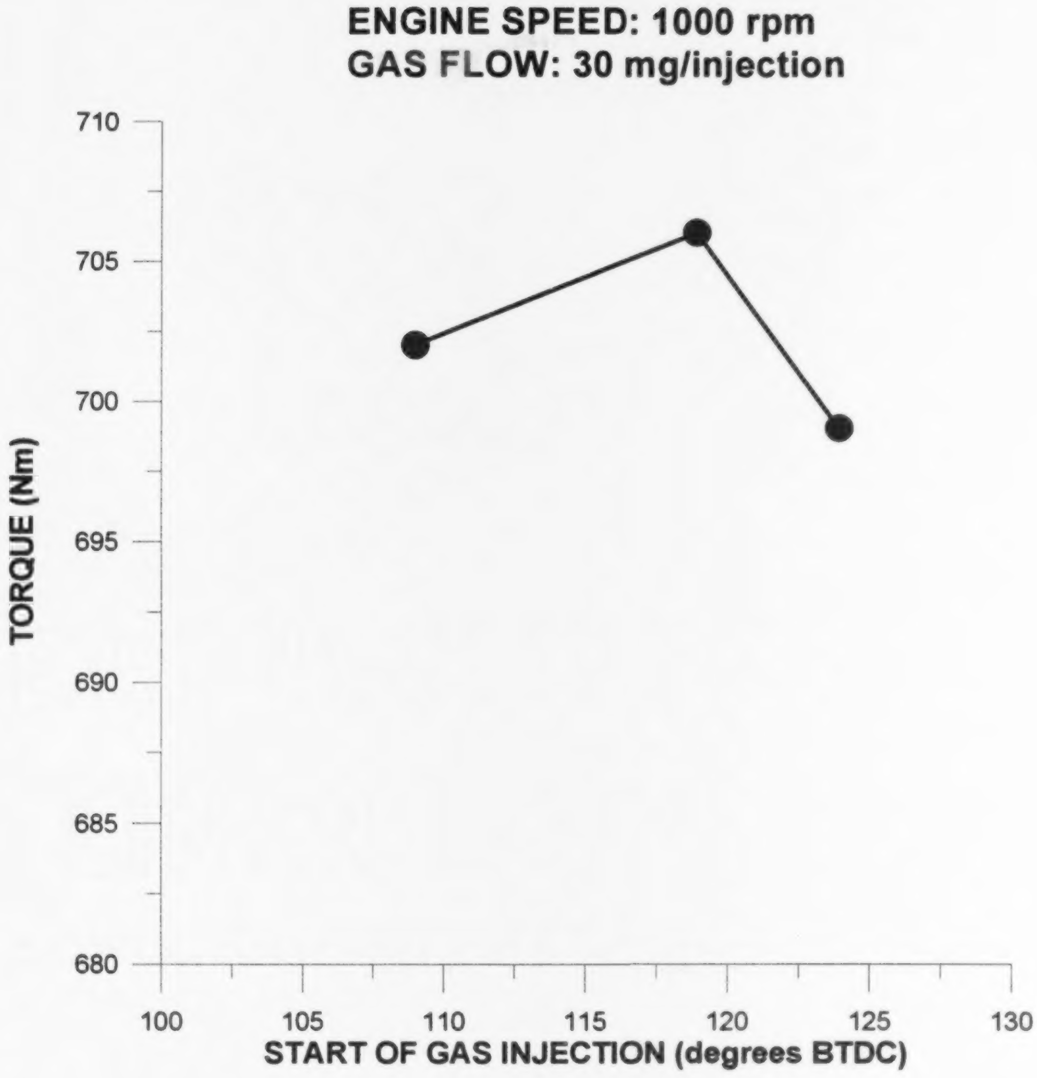


Figure 4.8 The effect of natural gas injection timing on engine torque at a constant speed and injection duration.

later for greater gas volumes. As the length of injection was increased for more power, the beginning-of-injection point moved earlier in the scavenging process. The natural gas injection timing maps for both types of delivery tubes are included in Appendix A.

4.2.4 Diesel Only and Dual Fuel Operation

After conversion, the engine is fully electronically controlled using a fail safe drive-by-wire system. A "throttle" position sensor provides a given torque demand and the software converts this to a fuel quantity in terms of equivalent mm^3 of diesel fuel per injection. During diesel only operation, the system looks up the required rack actuator position to supply this amount of diesel fuel from a speed/load rack actuator calibration table saved in the software. The pump lever position is continually adjusted to maintain desired fuel delivery as engine speed or torque demand varies. Fuel quantity is also affected by air density and is overridden by minimum and maximum speed governing.

Dual fuel operation is similar to diesel operation except that the system can split the total fuel quantity between natural gas and diesel fuel. To do this, the rack actuator position is set according to a diesel pilot map (also saved in the software). The pilot diesel quantity is then subtracted from the total fuel command, and the remainder is made up with natural gas to equivalent diesel fuel values. Natural gas injection timing is adjusted to the values stored in the gas valve timing map. Again, governing is provided at both programmed idle and maximum engine speed.

The engine is able to be readily switched from diesel only to dual fuel operation and back again, using a two-position switch on the control panel.

4.3 TESTING EQUIPMENT

The testing was conducted in the Engine Laboratory located in the Mechanical Engineering Building at the University of Alberta. The testing was carried out with the engine mounted on an eddy-current dynamometer which controlled engine speed by adjusting torque load. Engine torque output was controlled by an electronic "throttle" position controller which activated the "throttle input". Both engine speed and load were controlled by a pair of dedicated DyneSystems control computers: Digital Dynamometer Controller Dyn-Loc IV and Digital Throttle Controller DTC-1. Engine speed, power, and torque appeared on the digital readout and were simultaneously recorded by the computer data acquisition system. A water heat exchanger was used to keep the engine coolant and oil at normal operating temperatures. An additional cooler was installed on the diesel fuel return line to keep the diesel fuel temperature in the fuel tank at approximately 40°C. Keeping diesel fuel temperature constant allowed accurate control of the mass of injected diesel fuel.

Multiple measurements were continuously monitored and recorded during all tests. Dynamometer speed and torque were measured to calculate engine output power. A 220 litre air drum with two ASME long radius nozzles was used for air consumption measurement. The diameters of the nozzles were 73.60 mm and 63.59 mm respectively. The pressure difference between air drum and ambient pressure was measured by a Validyne type CD75 pressure transducer with 1380 Pa range. The air consumption was determined directly from the pressure difference according to SAE Recommended Practice J244. The calculation procedures are described in section 4.4. Supply tanks for both fuels, (diesel and natural gas), were continuously weighed by a pair of electronic scales. A straight line curve fit through the scale output was used to measure the fuel consumption rate. Chromel-alumel thermocouples were used to measure the exhaust gas temperatures in the tailpipe and at each of six exhaust ports. Diesel fuel, engine oil, engine coolant, air box, intake air dry bulb, and wet bulb temperatures were measured by copper-constantan thermocouples. Barometric pressure was obtained from a mercury barometer in the lab. A list of the equipment used for the measurement of engine torque and speed, fuel and air flow rates, and various temperatures is included in Table 4.3.

Table 4.3 List of the equipment used to measure engine torque, engine speed, fuel and air flow rates, and various temperatures.

Measured Quantity	Instrument Model	Range	Resolution
Engine Torque Engine Speed	Eddy-current Midwest Dynamometer Model 1014A	1000 N.m 10000 rpm	1 N.m 1 rpm
Fuel Flow Rate (Diesel fuel and natural gas)	Pacific Scale Model 550	0-60,000 g	5 g (~1g with averaging)
Air Flow Rate	Validyne Pressure Transducer Model CD75	0-1380 Pa	0.5 %
All temperatures except exhaust gas temperatures	Copper-Constantan Thermocouple	-270-600 °C	0.5 °C
Exhaust Gas Temperatures	Chromel-Alumel Thermocouple	-270-1370 °C	0.5 °C

The natural gas used for all tests came from the University of Alberta mains supply provided by Northwestern Utilities Limited. It was compressed from the building pressure to approximately 19 MPa (2760 psi) by a pair of "Fuelmaker" appliances and then stored in steel cylinders. The cylinders were generally refilled when the pressure had dropped to between 1/2 and 1/3 of full pressure in order to reduce gas supply variations. The composition of the natural gas used for all tests was presented earlier in Chapter 3 in Table 3.1. Esso summer diesel fuel (grade P-30) was used for all tests and its properties were presented in Table 3.2.

4.3.1 Exhaust Gas Analysis Equipment

The exhaust gas composition was measured continuously and recorded by various Engine Laboratory analyzers, giving readings of carbon dioxide (CO_2), carbon monoxide (CO), oxides of nitrogen (NO_x), total hydrocarbons (as hexane C_6H_{14}) (THC), and oxygen (O_2). A custom-built opacity meter (smokemeter) was installed on the 4 inch diameter exhaust pipe and used to measure soot emissions. This unit was calibrated according to SAE Recommended Practice J255a to give soot mass flow. A MTI (Microsensor Technologies Inc.) P200 gas chromatograph attached to a 80386-based PC-compatible computer was used to measure the exhaust gas composition on a sample-by-sample basis at each test point. The main purpose was to obtain methane concentration in the exhaust gas and to compare readings of carbon dioxide (CO_2) and carbon monoxide (CO) emissions with those measured by other analyzers.

The front panel readout and the voltage output of each analyzer were related to the concentration of pollutant in the exhaust. The zero reading of each analyzer was checked by supplying it with a pure nitrogen (N_2), which was used as a zero span gas. When necessary, the zero on the meter was adjusted to obtain a slightly positive, but close to zero, voltage output. Then miscellaneous span gases were supplied to the analyzer, and the concentration of each pollutant and the relevant voltage output were noted. Based on the concentration and voltage output various calibration equation were obtained using least squares curve fits. The CO_2, CO, O_2, and NO_x calibration equations were linear while that for the FID (THC) was a second-order polynomial. The gas chromatograph was calibrated for CO_2, CO, and CH_4 using a dedicated Windows software "Ezchrom". Photocell voltage output of the smokemeter is related linearly to the transmittance of the exhaust gases. Two dimmed slides, with the transmittance of 97.7 % and 93.3 % respectively, were used to calibrate the smokemeter. When no slide was used, an additional calibration point with the transmittance of 100 % was obtained. Details of the emission equipment and calibration are given in Table 4.4.

Figure 4.9 shows a schematic diagram of the exhaust gas analysis system. The exhaust gas was sampled directly from the tailpipe. The extracted exhaust gas was first cooled to dehumidify the sample gas, and then filtered to remove all particulate impurities. The dry and filtered exhaust sample was pumped to the gas analyzers. A set of control valves and flowmeters was used to maintain an appropriate pressure and flow rate to each instrument. The following analyzers were used: a Beckman 864 Non-dispersive Infrared (NDIR) analyzer to measure CO_2, a Beckman 864 NDIR analyzer to measure CO, a Beckman 955 Chemi-Luminescent analyzer to measure NO_x, a Taylor

Servomex OA.137 analyzer to measure O_2, a Beckman GC-72.5 Flame Ionization Detector (FID) to measure total hydrocarbons THC (as hexane C_6H_{14}), and a MTI P200 Gas Chromatograph to measure CH_4, CO, and CO_2.

Table 4.4 List of the emission measurement equipment.

Measured Pollutant	Model Serial	Range	Resolution	Calibration
CO_2	Beckman 864 NDIR	0-20 %	~ 0.05 %	0%, 9.9%, 15.1%
CO	Beckman 864 NDIR	0-1 %	~ 0.01 %	0%, 0.1%, 1.01%
HC	Beckman GC-72.5 Dual FID	0-3000 ppm	~ 1 ppm	0, 237, 477, 2067 ppm
NO_x	Beckman 955 Chemi-Luminescent	0-2500 ppm	~ 1 ppm	0, 242, 923 ppm
O_2	Taylor Servomex OA.137	0-25 %	~ 0.1 %	0, Room Air 21%
CH_4	MTI P200 GC	0-1,000,000 ppm	1 to 5 ppm	425,1425, 12400 ppm

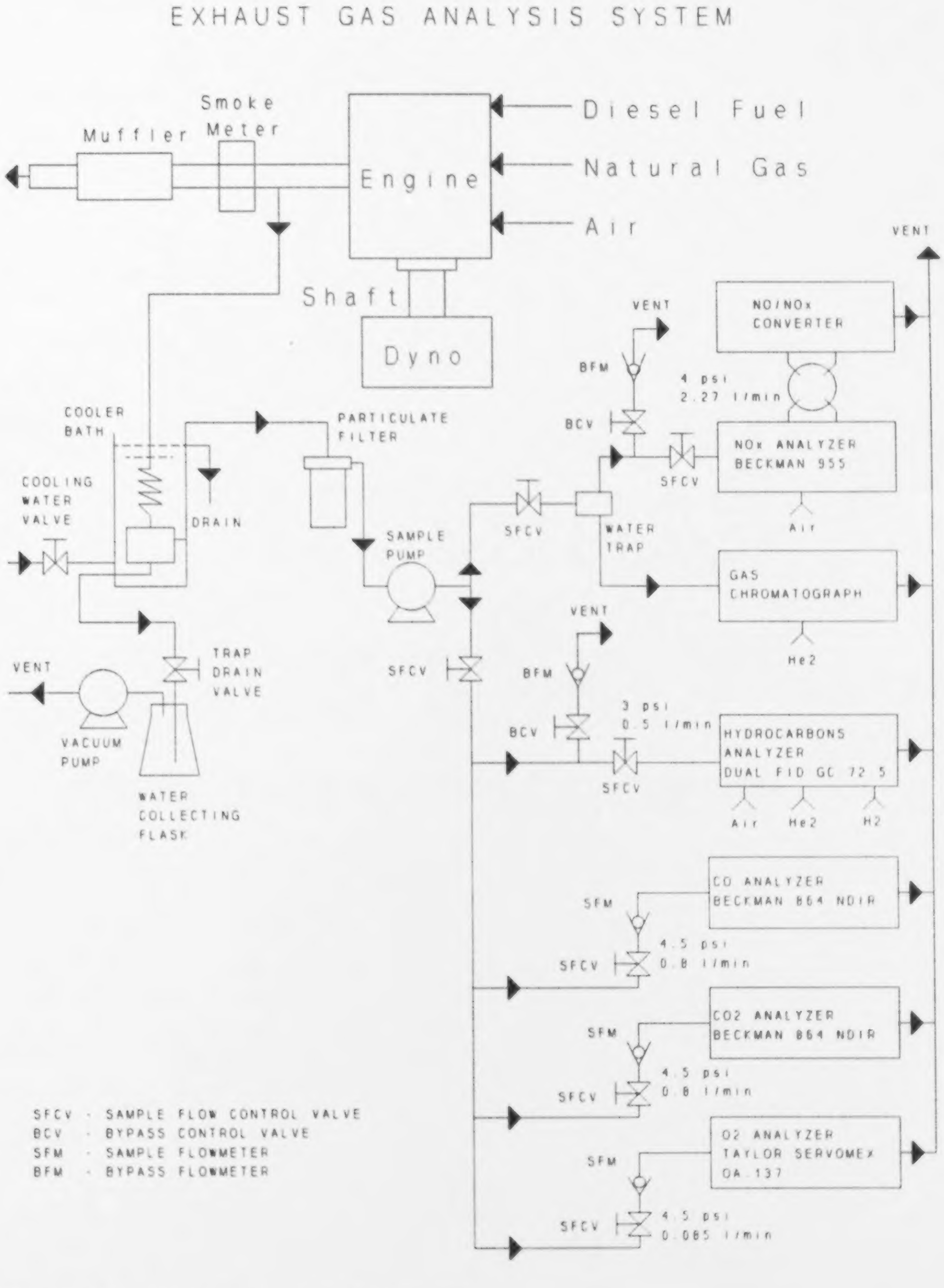

EXHAUST GAS ANALYSIS SYSTEM
Smoke Meter
Muffler
Engine
Diesel Fuel
Natural Gas
Air
Shaft
Dyno
VENT
NO/NOx CONVERTER
VENT
BFM
4 psi
2.27 l/min
BCV
NOx ANALYZER BECKMAN 955
SFCV
Air
COOLER BATH
PARTICULATE FILTER
COOLING WATER VALVE
DRAIN
SAMPLE PUMP
SFCV
WATER TRAP
GAS CHROMATOGRAPH
He2
VENT
BFM
TRAP DRAIN VALVE
VENT
SFCV
VACUUM PUMP
WATER COLLECTING FLASK
BCV
3 psi
0.5 l/min
HYDROCARBONS ANALYZER DUAL FID GC 72-5
SFCV
Air
He2
H2
SFM
CO ANALYZER BECKMAN 864 NDIR
4.5 psi
0.8 l/min
SFCV
SFM
CO2 ANALYZER BECKMAN 864 NDIR
4.5 psi
0.8 l/min
SFCV
SFCV - SAMPLE FLOW CONTROL VALVE
BCV - BYPASS CONTROL VALVE
SFM - SAMPLE FLOWMETER
BFM - BYPASS FLOWMETER
SFM
O2 ANALYZER TAYLOR SERVOMEX OA-137
SFCV
4.5 psi
0.085 l/min

Figure 4.9 Exhaust gas analysis system.

4.3.2 Data Acquisition Programs

The primary test variables were measured, displayed, and recorded by the dedicated data acquisition program running on an AT-compatible computer using a data acquisition and multiplexer cards. The data acquisition program "Busread" was written in Quickbasic programming language. The test measurements covered engine speed, engine torque, diesel fuel and natural gas consumption, air consumption, and exhaust gas composition as well as air temperature and relative humidity, exhaust gas temperatures, oil and coolant temperatures, and water bath temperature. "Busread" program was used for data acquisition, calibration and continuous strip chart display of multiple channels of data. It used VGA graphics (screen 12, 60 lines) for display and two Metrabyte boards (DAS-16G1 and EXP-16) for analog/digital conversion. The program used an input setup file containing information on how many channels to read, their names and units, and what calibration equation to apply. The information was then incorporated into the program and used throughout the test. All measured parameters were taken at one second intervals and continuously averaged over a thirty second period. The one second readings were continually recorded in a file as binary data. At each testing point or "mode", a thirty second average of the measured parameters was stored in an ASCII "modal" file. The two main measurements not monitored continually by the "Busread" program were the barometric pressure (measured at the start of each test) and the gas chromatograph readings of exhaust methane concentration (measured on a sample-by-sample basis by another 386-compatible computer).

"Busshow", another Quickbasic program, was used to read data from the binary output files produced by "Busread" for further review so that new modal files could be saved. "Busshow" practically performed the same functions as the "Busread" program; however, it used the recorded at one second intervals data as data input rather than reading live data.

4.4 TESTING PROCEDURE

4.4.1 Maximum Torque Test

During the maximum torque testing, the engine was run at maximum "throttle" for the following speeds: 800, 1000, 1200, 1400, 1600,1800, 2000, and 2200 rpm in increasing order. The main objective of the maximum torque test was to determine maximum torque, maximum power and brake specific fuel consumption over a wide range of speeds. Emissions results were also recorded. Although the maximum torque test differs from normal engine operation, the emissions results help to understand and to evaluate the performance of the dual fuel engine.

The engine was initially warmed up until the coolant temperature reached between 65°C and 75°C. All testing parameters including torque, power, fuel and air consumption as well as exhaust gas composition were measured at one second intervals by the computer. When conditions were stabilized, a thirty second average was stored in a file for further analysis. Two or more thirty second averages were saved at each tested engine speed, and were later averaged to get a representative value for each speed.

The maximum torque test was conducted first on diesel fuel with the fuel rack pushed fully forward. Then, for dual fuel operation, the Dual Fuel Injection System adjusted the fuel rack position to control the diesel pilot and actuated the gas injection solenoid valves to meter natural gas. During development, the natural gas replacement map was adjusted to be just below the point at which the audible engine knock would occur and the total fuel delivery (natural gas plus diesel fuel) was set to keep the maximum torque at approximately the same level as diesel operation. In the final testing, the system ran without adjustments using the stored maps. In addition, dual fuel tests were carried out with two different sets of gas injection tubes directing gas into the cylinder intake ports. One set of injection tubes pointed upwards (+30 deg) toward the exhaust valves , the other set pointed downwards (-30 deg) towards the piston surface.

4.4.2 The Japanese 13-Mode Test

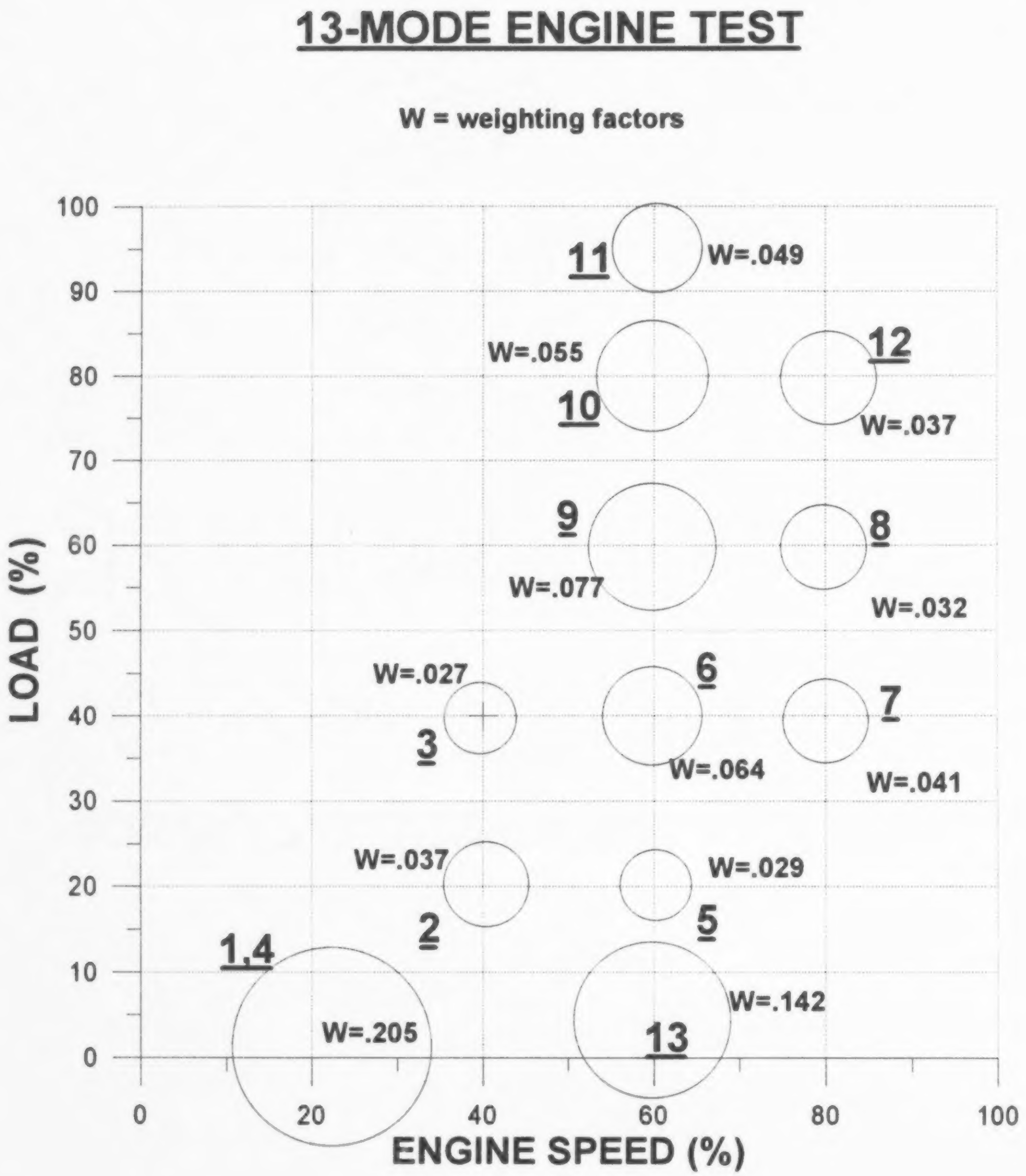
The 13-mode procedure was used to test the engine's emissions and fuel economy over a range of conditions typical for normal operation. The Japanese 13-mode procedure was recommended by Alternative Fuel Systems Inc., which has tested many four-stroke dual fuel engines on this cycle for export markets. The Japanese 13-mode test is an official standard procedure for testing diesel engines, so the fuel consumption and the emission rates over this test are intended to provide a standard evaluation over a typical range of operating conditions [18]. The testing procedure emphasizes idling and medium torque operation rather than just maximum torque operation. However, any other multi-mode cycle with a similar coverage would produce similar results.

For this test, the engine was run at steady-state conditions of speed and torque for each of thirteen modes. The operating points and weighting factors, (essentially operating time fractions) are shown in Table 4.5 and presented graphically in Figure 4.10. The weighting factors are used for calculating composite results from the individual modal values and they emphasize the operating conditions which are important in normal use. The three most heavily weighted modes are 1, 4 and 13; modes 1 and 4 represent engine idle and mode 13 is a very low load mode (only 5 % of full torque).

Table 4.5 The Japanese 13-mode test procedure.

Mode	% Rated Speed (Testing Speed)	% Full Load (Testing Load)	Weighting Factor
1	Idle (570 rpm)	0 % (0 N.m)	0.205
2	40 % (880 rpm)	20 % (144 N.m)	0.037
3	40 % (880 rpm)	40 % (288 N.m)	0.027
4	Idle (570 rpm)	0 % (0 N.m)	0.205
5	60 % (1320 rpm)	20 % (142 N.m)	0.029
6	60 % (1320 rpm)	40 % (284 N.m)	0.064
7	80 % (1760 rpm)	40 % (256 N.m)	0.041
8	80 % (1760 rpm)	60 % (384 N.m)	0.032
9	60 % (1320 rpm)	60 % (426 N.m)	0.077
10	60 % (1320 rpm)	80 % (568 N.m)	0.055
11	60 % (1320 rpm)	95 % (674 N.m)	0.049
12	80 % (1760 rpm)	80 % (512 N.m)	0.037
13	60 % (1320 rpm)	5 % (36 N.m)	0.142

As during the maximum torque test, the engine was tested for both 100 % diesel fuel operation and dual fuel operation. Modes 1, 4 and 13 were operated only on diesel



Note: Modes 1, 4 and 13 are operated only on diesel fuel.

Figure 4.10 The Japanese 13-mode test procedure.

fuel. In each mode of the Japanese 13-mode test, data of all measured parameters were taken at one second intervals by the computer and continuously averaged over a thirty second period. At least two files, containing all testing parameters, were saved for each mode as soon as the testing conditions were stabilized. They were averaged to get a representative value for each mode. The average results at each test mode were later multiplied by the appropriate weighting factor and summed to give the composite Japanese 13-mode test results.

4.5 DATA REDUCTION

After the tests were conducted, the recorded data were often reviewed using the "Busshow" program to ensure that the tests were conducted at appropriate conditions. The program was also used to save new modes, or to facilitate other manipulations necessary for further data deduction.

Two special programs were written in Quickbasic programming language for data reduction. "Maxload" performs data reduction for the maximum torque test results, while "13Mode" is dedicated for the 13-mode results. The "13Mode" program basically performs the same functions as the "Maxload" program; however, it carries additional calculations exclusively for the Japanese 13-mode test. These two programs essentially do the following :

1. Locate and read recorded data files for each tested speed or mode of a particular test.
2. Average all the recorded data for the repeats for a particular mode or speed to

obtain the representative values.

3. Correct torque and power output according to the procedure described in SAE J1349 [5].

4. Calculate the air consumption, fuel consumption for diesel and natural gas, and emissions in g/s and in g/kW.h.

Additionally, the "13Mode" program uses the mode weighting factors from Table 4.5 and calculates the composite values for the emissions and fuel flow rates in g/kW.h for the Japanese 13-mode test. For this calculation, mode emissions and fuel flow rates are calculated in g/s, multiplied by the mode weighting factor, and summed to form a numerator. Mode powers are multiplied by the mode weighting factor and summed to give a denominator. The resultant ratio gives a weighted fuel consumption and emission value in specific form: g/kW.h.

4.5.1 Calculation of Air Intake Flow.

As described earlier, an air drum equipped with two ASME long radius nozzles was used for air flow measurement. The pressure difference between the air drum and ambient pressure was measured and used by the data reduction programs to calculate the air intake flow rate. The calculations of the intake airflow presented below follow SAE Recommended Practice J244 procedure [6].

Air viscosity (in cP) is calculated according to equation 4.1, where T_{dry} is a known dry bulb temperature in °C :

Equation 4.2 converts the air viscosity from cP to $N.s/m^2$:

$$\mu_{Air}\ (cP) = 0.01714153 + 4.873549\times10^{-5}\times(T_{dry}) - 2.798293\times10^{-8}\times(T_{dry})^2 \qquad \textbf{(4.1)}$$

$$\mu_{Air}(Ns/m^2) = \mu_{Air}(cP)\ /\ 1000 \qquad \textbf{(4.2)}$$

Air density (kg/m^3) is determined by equation 4.3, where atmospheric pressure (P_{atm} in kPa), molar mass of air (M_{Air} in kg/kmol), ideal gas constant (R), and dry bulb temperature (T_{dry} in °C) are known variables.

$$\rho_{Air}(kg/m^3) = (P_{atm} \times M_{Air})\ /\ (R \times T_{dry}) \qquad \textbf{(4.3)}$$

Air kinematic viscosity (in m^2/s) is obtained from equation 4.4 :

$$\nu_{Air}(m^2/s) = \mu_{Air}(Ns/m^2)\ /\ \rho_{Air} \qquad \textbf{(4.4)}$$

Velocity of the air flowing through the ideal nozzle is calculated with equation 4.5, based on Bernoulli obstruction meter theory. The pressure difference (dP in Pa) is a measured variable.

$$V(m/s) = \sqrt{(2\times0.98\times dP)\ /\ \rho_{Air}} \qquad \textbf{(4.5)}$$

Hence, Reynolds number (Re) for assumed ideal nozzle of diameter D_{nozzle} is :

$$Re = (V\times D_{nozzle})\ /\ \nu_{Air} \qquad \textbf{(4.6)}$$

The coefficient of discharge (Cd) for the long radius nozzle is determined from the following equation [6]:

$$Cd = 0.19436 + 0.152884\times\ln Re - 0.0097785\times(\ln Re)^2 + 2.0903\times10^{-4}\times(\ln Re)^3 \qquad \textbf{(4.7)}$$

Finally, the volumetric and mass flow rates can be determined from equations 4.8 and 4.9 respectively :

$$Q(m^3/s) = Cd \times \frac{\Pi \times (D_{nozzle})^2}{4} \times \sqrt{\frac{(2 \times dP)}{\rho_{Air}}} \tag{4.8}$$

$$\dot{m}(kg/s) = Q \times \rho_{Air} = Cd \times \frac{\Pi \times (D_{nozzle})^2}{4} \times \sqrt{2 \times dP \times \rho_{Air}} \tag{4.9}$$

4.5.2 Calculation of Exhaust Composition Molar Fractions on Dry Basis

Based on the measured exhaust gas concentrations, the programs calculate the molar fractions of the measured components in the dry exhaust gas:

$$yCO_{dry} = CO_{per} \;/\; 100 \tag{4.10}$$

$$yCO_{2\;dry} = CO_{2\;per} \;/\; 100 \tag{4.11}$$

$$yO_{2\;dry} = O_{2\;per} \;/\; 100 \tag{4.12}$$

$$yNO_{dry} = NO_{x\;ppm} \;/\; 1000000 \tag{4.13}$$

$$yCH_{4\;dry} = CH_{4\;ppm} \;/\; 1000000 \tag{4.14}$$

where CO_{per}, $CO_{2\ per}$, $O_{2\ per}$, $NO_{x\ ppm}$, and $CH_{4\ ppm}$ are the concentrations of carbon monoxide, carbon dioxide, oxygen, oxides of nitrogen, and methane in the dry exhaust respectively. The molar fractions of carbon monoxide, carbon dioxide, oxygen, oxides of nitrogen, and methane are identified as yCO_{dry}, $yCO_{2\ dry}$, $yO_{2\ dry}$, yNO_{dry}, and $yCH_{4\ dry}$ respectively.

The total hydrocarbons FID reading is approximated as hexane and the program subtracts off the methane composition to obtain the reactive hydrocarbons (RHC) value:

$$yTHC_{dry} = THC_{ppm} \ / \ 1000000 \qquad \textbf{(4.15)}$$

$$yRHC_{dry} = yTHC_{dry} - yCH_{4\ dry} \ / \ 6 \qquad \textbf{(4.16)}$$

where THC_{ppm} is the concentration of total hydrocarbons, while $yTHC_{dry}$ and $yRHC_{dry}$ are the molar fractions of total hydrocarbons and reactive hydrocarbons in the dry exhaust gas, respectively.

The saturated vapour pressure P_{sat} (in kPa) at the cooler bath temperature is calculated:

$$P_{sat} = \exp(c_1/T_{bath}+c_2+c_3\times T_{bath}+c_4\times T_{bath}^{\ 2}+c_5\times T_{bath}^{\ 3}+c_6\times \ln T_{bath}) \ / \ 1000 \qquad \textbf{(4.17)}$$

Where c_1, c_2, c_3, c_4, c_5, and c_6 are constants, and T_{bath} is the measured cooler bath temperature in K [50]. The mole fraction of the water vapour yH_2O_{dry} in the "dry" exhaust gas flowing through the gas analyzers is then approximated:

$$yH_2O_{dry} = P_{sat} \ / \ P_{barom} \qquad \textbf{(4.18)}$$

where P_{barom} is the barometric pressure in kPa.

The mole fraction of nitrogen $yN_{2\ dry}$ is the remainder of the molar emission flow rate:

$$yN_{2\ dry} = 1-(yCO_{dry}+yCO_{2\ dry}+yO_{2\ dry}+yNO_{dry}+yRHC_{dry}+yCH_{4\ dry}+yH_2O_{dry}) \qquad \textbf{(4.19)}$$

4.5.3 Determination of Unknown Emission Component Molar Fractions

The combustion equation for a hydrocarbon fuel in air is given by:

$$C_xH_y+fO_2+3.76fN_2 \rightarrow aCO_2+bO_2+dCO+eN_2+gH_2O+jC_6H_{14}+kNO \tag{4.20}$$

Where a, b, d, e, j, and k are the known coefficients, while f, x, y, g are the unknown coefficients.

The principle of conservation of mass is used to solve the unknown coefficients.

Carbon balance:

$$x = yCO_{dry}+yCO_{2\ dry}+6\times yRHC_{dry}+yCH_{4\ dry} \tag{4.21}$$

Nitrogen balance:

$$f = (yN_{2\ dry}+0.5\times yNO_{dry})\ /\ 3.76 \tag{4.22}$$

Oxygen balance:

$$g = 2\times(f-yCO_{2\ dry}-yO_{2\ dry}-0.5\times yCO_{dry}-0.5\times yNO_{dry}) \tag{4.23}$$

Hence, the molar fraction of water vapour (from the combustion process) is obtained from:

$$yH_2O^{*}_{dry} = g \tag{4.24}$$

Hydrogen balance:

$$y = 2\times g+14\times yRHC_{dry}+4\times yCH_{4\ dry} \tag{4.25}$$

Hence, the hydrogen to carbon ratio (H/C) calculated from emissions is:

$$(H\ /\ C)_{em} = \frac{y}{x} \tag{4.26}$$

Air-fuel ratio derived from emissions $(A/F)_{em}$ is:

$$AF_{em} = 4.76 \times f \times M_{Air} \ / \ (12 \times x + y) \tag{4.27}$$

and air-fuel ratio measured directly from mass flow rates is:

$$AF_{mass} = \frac{Air\ Flow_{mass}}{Diesel\ Fuel\ Flow_{mass} + Natural\ Gas\ Flow_{mass}} \tag{4.28}$$

4.5.4 Calculation of Molar Fractions on Wet Basis

The mole fraction of water vapour is added to the dry exhaust and the composition of the wet exhaust gas prior to a water trap is recalculated to include water vapour:

$$Exh_{wet} = 1 + yH_2O^{*}_{dry} \tag{4.29}$$

$$yCO_{2\ wet} = yCO_{2\ dry} \ / \ Exh_{wet} \tag{4.30}$$

$$yCO_{wet} = yCO_{dry} \ / \ Exh_{wet} \tag{4.31}$$

$$yO_{2\ wet} = yO_{2\ dry} \ / \ Exh_{wet} \tag{4.32}$$

$$yNO_{wet} = yNO_{dry} \ / \ Exh_{wet} \tag{4.33}$$

$$yTHC_{wet} = yTHC_{dry} \ / \ Exh_{wet} \tag{4.34}$$

$$yRHC_{wet} = yRHC_{dry} \ / \ Exh_{wet} \tag{4.35}$$

$$yCH_{4\ wet} = yCH_{4\ dry} \ / \ Exh_{wet} \tag{4.36}$$

$$yN_{2\,wet} = yN_{2\,dry} \,/\, Exh_{wet} \tag{4.37}$$

$$yH_2O_{wet} = yH_2O^{*}_{dry} \,/\, Exh_{wet} \tag{4.38}$$

where yCO_{2wet}, yCO_{wet}, yO_{2wet}, yNO_{wet}, $yTHC_{wet}$, $yRHC_{wet}$, yCH_{4wet}, yN_{2wet}, and yH_2O_{wet} are the molar concentrations of carbon dioxide, carbon monoxide, oxygen, oxides of nitrogen, total hydrocarbons, reactive hydrocarbons, methane, nitrogen, and water vapour in the total (on a wet basis) exhaust gas, respectively.

The molar mass of the exhaust gas, M_{exh}, is calculated by multiplying the molar fraction of each exhaust gas component on wet basis by its molar mass, and adding the products together:

$$\begin{aligned} M_{exh} = {} & 44.01 \times yCO_{2\,wet} + 28.01 \times yCO_{wet} + 31.99 \times yO_{2\,wet} + 30 \times yNO_{wet} \\ & + 86 \times yTHC_{wet} + 28.01 \times yN_{2\,wet} + 18.01 \times yH_2O_{wet} \end{aligned} \tag{4.39}$$

The total exhaust mass flow rate is calculated as follows:

$$Exhtot = Air\ Flow_{mass} + Diesel\ Fuel\ Flow_{mass} + Natural\ Gas\ Flow_{mass} \tag{4.40}$$

where Exhtot is the total exhaust flow rate in g/s.

4.5.5 Calculation of Mass Flow Rate of Components

The mass flow rates of the individual components, in grams per second, are calculated from the total exhaust mass flow and the molar fractions on a wet basis:

$$mCO_{2\,gps} = yCO_{2\,wet}\times Exhtot\times 44.01\ /\ M_{exh} \tag{4.41}$$

$$mCO_{gps} = yCO_{wet}\times Exhtot\times 28.01\ /\ M_{exh} \tag{4.42}$$

$$mO_{2\,gps} = yO_{2\,wet}\times Exhtot\times 31.99\ /\ M_{exh} \tag{4.43}$$

$$mNO_{x\,gps} = yNO_{wet}\times Exhtot\times(\frac{46}{30})\times 30\ /\ M_{exh} \tag{4.44}$$

$$mRHC_{gps} = yRHC_{wet}\times Exhtot\times 86\ /\ M_{exh} \tag{4.45}$$

$$mCH_{4\,gps} = yCH_{4\,wet}\times Exhtot\times 16.04\ /\ M_{exh} \tag{4.46}$$

$$mN_{2\,gps} = yN_{2\,wet}\times Exhtot\times 28.01\ /\ M_{exh} \tag{4.47}$$

$$mH_2O_{gps} = yH_2O_{wet}\times Exhtot\times 18.01\ /\ M_{exh} \tag{4.48}$$

where mCO_{2gps}, mCO_{gps}, mO_{2gps}, mNO_{xgps}, $mRHC_{gps}$, mCH_{4gps}, mN_{2gps}, and mH_2O_{gps} are the mass flow rates (in g/s) of carbon dioxide, carbon monoxide, oxygen, oxides of nitrogen, reactive hydrocarbons, methane, nitrogen, and water vapour in the wet exhaust gas, respectively.

When the engine is operated only on diesel fuel, the total hydrocarbons mass flow rate is calculated as follows:

$$mTHC_{gps} = yTHC_{wet}\times Exhtot\times 86\ /\ M_{exh} \tag{4.49}$$

where $mTHC_{gps}$ is the mass flow rate (in g/s) of total hydrocarbons in the wet exhaust gas.

During dual fuel operation, methane is the dominant component of the total hydrocarbons emissions and the mass flow rate is determined as follows:

$$mTHC_{gps} = yTHC_{wet} \times Exhtot \times 6 \times 16.04 \; / \; M_{exh} \tag{4.50}$$

4.5.6 Calculation of the Brake Specific Exhaust Emissions and Fuel Flow Rates

The brake specific exhaust emissions are determined directly from the emission mass flow rates in grams per second (g/s) by multiplying by 3600 seconds and then dividing by the brake power, $\text{Power}_{\text{brake}}$, in kW. The results are thus presented in grams per kilowatt hour (g/kW.h):

$$bsCO_{2\;g/kWh} = mCO_{2\;gps} \times 3600 \; / \; Power_{brake} \tag{4.51}$$

$$bsCO_{g/kWh} = mCO_{gps} \times 3600 \; / \; Power_{brake} \tag{4.52}$$

$$bsO_{2\;g/kWh} = mO_{2\;gps} \times 3600 \; / \; Power_{brake} \tag{4.53}$$

$$bsNO_{x\;g/kWh} = mNO_{x\;gps} \times 3600 \; / \; Power_{brake} \tag{4.54}$$

$$bsRHC_{g/kWh} = mRHC_{gps} \times 3600 \; / \; Power_{brake} \tag{4.55}$$

$$bsCH_{4\;g/kWh} = mCH_{4\;gps} \times 3600 \; / \; Power_{brake} \tag{4.56}$$

$$bsN_{2\;g/kWh} = mN_{2\;gps} \times 3600 \; / \; Power_{brake} \tag{4.57}$$

$$bsH_2O_{g/kWh} = mH_2O_{gps} \times 3600 \ / \ Power_{brake} \tag{4.58}$$

$$bsTHC_{g/kWh} = mTHC_{gps} \times 3600 \ / \ Power_{brake} \tag{4.59}$$

where $bsCO_{2\,g/kWh}$, $bsCO_{g/kWh}$, $bsO_{2\,g/kWh}$, $bsNO_{x\,g/kWh}$, $bsRHC_{g/kWh}$, $bsCH_{4\,g/kWh}$, $bsN_{2\,g/kWh}$, $bsH_2O_{g/kWh}$, and $bsTHC_{g/kWh}$ are the brake specific emission rates of carbon dioxide, carbon monoxide, oxygen, oxides of nitrogen, reactive hydrocarbons, methane, nitrogen, water vapour, and total hydrocarbons, respectively.

In a similar way, the brake specific consumption of diesel fuel and natural gas are calculated from the corresponding fuel mass flow rates. They are also presented in g/kW.h:

$$bsfc_{Diesel} = Diesel\ Fuel\ Flow_{mass} \times 3600 \ / \ Power_{brake} \tag{4.60}$$

$$bsfc_{NatGas} = Natural\ Gas\ Flow_{mass} \times 3600 \ / \ Power_{brake} \tag{4.61}$$

The brake specific natural gas consumption is converted to energy equivalent of diesel fuel as follows:

$$bsfc_{NatGasDE} = bsfc_{NatGas} \times \frac{LHV_{NatGas}}{LHV_{Diesel}} \tag{4.62}$$

where $bsfc_{NatGasDE}$ is a diesel equivalent of the brake specific natural gas consumption, while LHV_{Diesel}, and LHV_{NatGas} represent the lower heating values in kilojoules per gram (kJ/g) of diesel fuel and natural gas, respectively. The following lower heating values are used:

LHV_{Diesel} = 43.170 kJ/g for diesel fuel [51], and

LHV_{NatGas} = 45.268 kJ/g for natural gas (based on the natural gas composition and LHV of each component [4]).

4.5.7 Calculation of Soot Emissions

As described earlier, a custom-built full flow opacity meter was installed on the 4 inch diameter exhaust pipe and used to determine soot emissions. The instrument is calibrated to give a light transmittance reading, expressed in percent (%). The light transmittance reading is then converted to opacity (also expressed in %):

$$Opacity\ (\%) = 100 - Transmittance\ (\%) \tag{4.63}$$

The corresponding exhaust soot concentration is determined from equation 4.64, which is derived from the correlation between exhaust gas opacity and exhaust soot content for 4 inch stack as described in SAE Recommended Practice J255a [7].

$$Soot_{gpkg} = 0.01418 \times Opacity\ (\%) \tag{4.64}$$

The exhaust soot concentration, $soot_{gpkg}$, is expressed in grams of soot per kilogram of exhaust gas. In order to obtain a soot mass flow rate, the exhaust soot concentration is multiplied by the total exhaust mass flow rate in g/s and divided by 1000:

$$mSoot_{gps} = Soot_{gpkg} \times Exhtot\ /\ 1000 \tag{4.65}$$

where $mSoot_{gps}$ is the soot mass flow rate in g/s.

As in the case of other emissions, the brake specific soot flow rate, $bsSoot_{g/kWh}$, is calculated from the soot mass flow rate in g/s by multiplying by 3600 seconds and then dividing by the brake power:

$$bsSoot_{g/kWh} = mSoot_{gps} \times 3600 \ / \ Power_{brake} \tag{4.66}$$

4.5.8 Thermal Efficiency Calculation

The thermal efficiency, η_T, is defined as the ratio of the net work out to the total fuel energy supplied to the engine. In this study, the thermal efficiency calculation is based on the engine's brake power and on the fuel's lower heating value (LHV):

$$\eta_T(\%) = \frac{Power_{brake} \times 100\%}{Diesel\ Fuel\ Flow_{mass} \times LHV_{Diesel} + Natural\ Gas\ Flow_{mass} \times LHV_{NatGas}} \tag{4.67}$$

where LHV_{Diesel}, and LHV_{NatGas} are the lower heating values in kilojoules per gram (kJ/g) of diesel fuel and natural gas, respectively.

4.5.9 Diesel Fuel Substitution Calculation

Diesel fuel substitution is defined as the amount of diesel fuel replaced by natural gas in dual fuel operation compared to the amount of diesel fuel needed to produce the same load at the same engine speed in 100% diesel operation. Diesel fuel substitution is calculated as follows:

$$DFSub\ (\%) = \left(1 - \frac{(Diesel\ Fuel\ Flow_{mass})_{Dual\ Fuel}}{(Diesel\ Fuel\ Flow_{mass})_{Diesel\ Only}}\right) \times 100\% \tag{4.68}$$

where DFSub is the diesel fuel substitution (replacement) expressed in percent (%). For example, 40 % substitution means that for given speed and load dual fuel operation uses 60 % of the diesel fuel which would be consumed in diesel only operation.

4.5.10 Composite Results for the Japanese 13-Mode Test

A single composite value for each of the exhaust emission components and for each fuel consumed is calculated in g/kW.h over the 13 test modes. The composite brake specific emissions and the composite brake specific fuel consumptions are determined in the following way:

1. The average emission and fuel mass flow rates in g/s at each test mode are multiplied by the appropriate weighting factors and 3600 seconds. The resulting products are then summed to obtain the total weighted emission and fuel mass flow rates in grams per hour (g/h).
2. The average power outputs in kW at each mode are multiplied by the appropriate weighting factors. The resulting products are summed to obtain the total weighted power output in kW.
3. Finally, the composite brake specific emissions and the composite brake specific fuel consumptions are calculated by dividing the total weighted emission and fuel mass flow rates by the total weighted power output.

The mode weighting factors in table 4.5 are used to calculate the composite results. For example, the composite brake specific carbon dioxide emissions over the Japanese 13-mode test are determined from the following equation:

$$BSCO_2 = \frac{\sum_{n=1}^{13} (CO_{2_{g/s}} \times \text{Weighting factor})}{\sum_{n=1}^{13} (Power_{brake} \times \text{Weighting factor})} \frac{g}{kWh} \qquad \textbf{(4.69)}$$

The composite brake specific emissions of carbon monoxide (BSCO), oxides of nitrogen

(BSNO$_x$), total hydrocarbons (BSTHC), reactive hydrocarbons (RHC), methane (BSCH$_4$), and soot (BSSoot) are calculated in a similar manner. Likewise, the composite brake specific fuel consumptions for diesel and natural gas fuel are determined from equations 4.70 and 4.71, respectively:

$$BSFC_{Diesel} = \frac{\sum_{n=1}^{13} (Diesel\ Fuel\ Flow_{mass} \times Weighting\ factor)}{\sum_{n=1}^{13} (Power_{brake} \times Weighting\ factor)} \frac{g}{kWh} \qquad \textbf{(4.70)}$$

$$BSFC_{NatGas} = \frac{\sum_{n=1}^{13} (Natural\ Gas\ Flow_{mass} \times Weighting\ factor)}{\sum_{n=1}^{13} (Power_{brake} \times Weighting\ factor)} \frac{g}{kWh} \qquad \textbf{(4.71)}$$

4.5.11 Calculation of Corrected Power and Torque

The performance of a compression ignition type engine is affected by the density of the inlet combustion air. To provide a common basis of comparison, it is necessary to apply a correction factor to the observed power to account for differences between the reference air conditions and those at which test data were acquired. For the maximum torque test results, a dedicated program corrects all maximum power and torque results to standard atmospheric conditions according to SAE Standard J1349 [5].

$$Power_{corrected} = Air_{factor} \times Power_{observed} \qquad \textbf{(4.72)}$$

The corrected torque Torque$_{corrected}$ is calculated directly from the corrected power Power$_{corrected}$:

$$Torque_{corrected} = \frac{60\ 000 \times Power_{corrected}}{2 \times \Pi \times Engine\ speed} \tag{4.73}$$

The atmospheric correction factor Air_{factor} is described by SAE J1349 as follows:

$$Air_{factor} = (fa)^{fm} \tag{4.74}$$

The value of atmospheric factor fa is given as:

$$fa = (99\ /\ P_{do}) \times (T_{dry}\ /\ 298)^{0.7} \tag{4.75}$$

where P_{do} is the inlet manifold total pressure (refers to dry air portion), kPa

$$P_{do} = \frac{P_{barom}}{760} \times 101.325 - P_v \tag{4.76}$$

and

T_{dry} = Inlet air supply temperature (dry bulb), K

P_{barom} = Barometric pressure, mm Hg

P_v = Atmospheric vapour pressure, kPa

The value of engine factor fm is given as:

0.2	for q less than 37.2
$(0.036 \times q)$-1.14	for q between 37.2 and 65, or
1.2	for q higher than 65

Where the fuel delivery q (in mg/L cycle) is calculated as:

$$q = \frac{60\ 000 \times (Fuel\ Flow_{mass})}{Engine\ displacement \times Engine\ speed} \tag{4.77}$$

4.6 REPEATABILITY AND ACCURACY OF MEASUREMENTS AND CALCULATED VALUES

Engine-based measurements are quite complex and dependent on many different factors including the calibration accuracy and the resolution of each measuring instrument. It is very important that each measurement can be repeated within acceptable accuracy. Therefore, it is necessary to determine the uncertainty of individual measurements and of calculated quantities.

All of the tested parameters were recorded as 30 second time averages in order to reduce variability of the measurement. At least two (very often three, four, or more) repeat measurements (30 second averages) were taken at each testing engine speed of the maximum torque test and at each mode of the Japanese 13-mode test. The variability between these measurements provides a good estimate of experimental uncertainty associated with each measurement and with the calculated composite 13-mode test results.

Relative standard deviation, defined as a ratio of the standard deviation of the sample to the mean value, was calculated between repeated emission and fuel consumption measurements at single test points. It was determined for both testing procedures in diesel only and dual fuel mode, and typical values are presented in Table 4.6.

The variabilities of all measured and calculated parameters are generally very low. Due to a digital accuracy of the dynamometer controllers, it was possible to run the engine exactly at test points with a minimal drift in engine speed and load. The maximum relative deviation of engine speed was always below 1.0 % while that of engine load did not exceed 0.6 % and 3.1 % for the maximum torque test and the 13-mode test, respectively.

Table 4.6 Relative standard deviation in repeat measurements at a single test point for both testing procedures in diesel only and dual fuel mode. (± 1 standard deviation, averages for all tested speeds or modes).

Measured Parameter	Maximum Torque Test		13-Mode Test	
	Diesel Only	Dual Fuel	Diesel Only	Dual Fuel
O_2	0.19 %	0.53 %	0.17 %	0.18 %
CO	2.53 %	4.34%	1.95 %	5.52%
CO_2	0.11 %	0.35 %	0.35 %	0.34 %
NO_x	0.55 %	3.57 %	3.91 %	1.03 %
THC	10.02 %	1.16 %	6.43 %	2.20 %
Transmittance	0.02 %	0.06 %	0.01 %	0.03 %
Diesel Consumption	1.89 %	1.35 %	3.86 %	4.07 %
NG Consumption	N/A	1.74 %	N/A	2.31 %

Oxygen, and carbon dioxide are major components of the exhaust mixture (percent rather than parts per million) so their measurements need to be precise. The maximum average relative deviations were found to be 0.53 % for oxygen and 0.35 % for carbon dioxide. The variabilities of CO, NO_x, and THC were significantly higher (0.55 to 10.02 %) mainly because small absolute deviations give relatively high relative deviations for the low levels of emissions. This relationship can be clearly seen in the case of the total hydrocarbons emissions. The relative deviation was 10.02 % during the maximum torque test in diesel only mode, where the THC emissions were practically near zero (less than 20 ppm as hexane). In contrast, the relative deviation dropped to 1.16 % when the engine was operated in dual fuel mode and the THC emissions were much higher (277-2525 ppm as hexane). The engine's exhaust gas was very clear and its transmittance was close to 100 % for most of the operating conditions. This resulted in a very low variability of the transmittance measurement (0.01-0.06 %). The relative deviation in fuel consumption rate was low, ranging from 1.35 to 1.89 % during the maximum torque tests where high consumption rates were measured. During the 13-mode tests, the variability of the fuel consumption measurement increased to 4.07 %. The high uncertainty in diesel consumption measurements is attributed to a high recirculating flow back to the tank. When engine load (and fuel consumption) is low, slight variations in the fuel contained in the engine and lines cause a high relative standard deviation.

The final results for the 13-mode test are based on weighted averages of the individual measurements in 13 different modes. Therefore, the variability of the 13-mode test results can be approximated by multiplying the variability of a single measurement by $1/\sqrt{13} = 0.277$. Table 4.7 presents the uncertainty of the 13-mode results in diesel only and dual fuel mode. The measurement uncertainty in the 13-mode composite results is very low and never exceeds 1.78 %. However, certain factors such as changes in the accuracy or calibration of the instruments, changes in the ambient temperature and pressure, changes in the coolant and lubricant temperatures, or other parameters would tend to increase the uncertainty for the test results obtained in this study.

Table 4.7 Relative standard deviation of the 13-mode test results in diesel only and dual fuel mode (± 1 standard deviation on repeatability analysis of single measurements in 13-mode test).

Measured Parameter	13-Mode Test	
	Diesel Only	Dual Fuel
O_2	0.05 %	0.05 %
CO	0.54 %	1.53 %
CO_2	0.10 %	0.09 %
NO_x	1.08 %	0.28 %
THC	1.78 %	0.61 %
Transmittance	0.003 %	0.008 %
Diesel Consumption	0.51 %	0.52 %
NG Consumption	N/A	0.64 %

CHAPTER 5 RESULTS AND DISCUSSION

The DDC 6V-71 diesel engine was converted to dual fuel operation with the AFS Electronic Dual Fuel Injection System. After the conversion, the engine was able to run in both diesel only and dual fuel modes. A baseline test was conducted on the original engine over the maximum torque test and the Japanese 13-mode test procedures. The converted engine was tested over the same testing procedures in diesel only and dual fuel mode. Also, two different sets of gas injection tubes were employed when the engine operated in dual fuel mode and tests were repeated with both sets.

The main objective of the maximum torque test was to determine the maximum torque, maximum power, and brake specific fuel consumption over the full engine speed range. Although the emission results obtained during maximum torque testing are not relevant to normal engine operation, they are very helpful during the process of evaluation of the dual fuel engine. Thus, the brake specific emissions measured during the maximum torque test are also presented in Section 5.1 of this chapter. Appendix B includes emissions and fuel consumption results as well as the raw experimental results obtained in the maximum torque tests.

During the Japanese 13-mode test, the emissions and fuel consumption rates were measured in each mode. These modal values were then combined into a set of composite values using appropriate weighting factors. The brake specific emissions and fuel consumption rates together with the composite results for the multi-mode procedure are presented in Section 5.2 of this chapter. Appendix C includes emissions, fuel consumption, and the composite results as well as the raw experimental results obtained in the 13-mode tests.

5.1 MAXIMUM TORQUE TEST RESULTS

5.1.1 Brake Power and Brake Torque

All the brake power and torque results were corrected to standard atmospheric conditions according to SAE J1349 [5]. The power correction factors varied from 2% to 5% and fell within the acceptable range of 5 % specified by SAE J1349.

As shown in Figure 5.1, the converted engine, when operated in diesel only or in dual fuel mode matches the power output of the stock engine very well. For all configurations, the shape of the power/speed curves is almost identical, showing a smooth increase of power with engine speed. For baseline diesel and dual fuel operation with tubes pointed down, the peak power of 131 kW and 136 kW was measured at 2200 rpm, respectively. A small decrease of power was observed above 2000 rpm for diesel only operation after conversion and for dual fuel operation with tubes pointed up. The peak power of 126.9 kW and of 128.5 kW was produced at 2000 rpm for these configurations.

The conversion had a little effect on the engine's torque output. It is clear from Figure 5.1 that the torque/speed curves of the converted engine operated either on straight diesel or on dual fuel follow virtually the same shape as for baseline diesel operation. However, slightly lower torque levels were observed after conversion over most of the speed range. For all test configurations, the peak torque was obtained at 1000 rpm and ranged between 736.5 N.m and 758 N.m. The highest peak torque (758 N.m) was

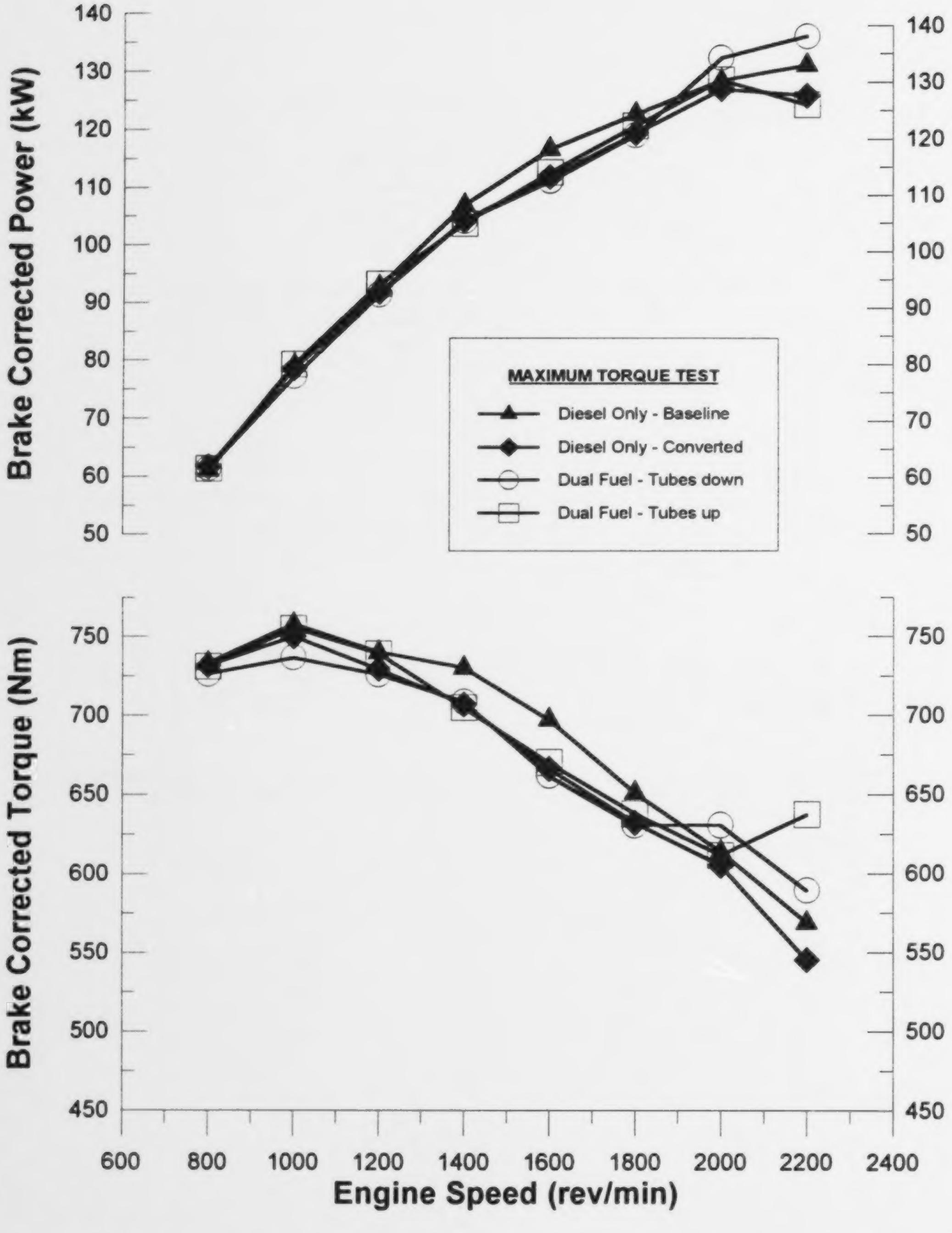


Figure 5.1 Corrected brake power and torque results for diesel only and dual fuel operation at maximum torque.

obtained from baseline diesel operation, while the lowest value (736.5 N.m or 3% less) was produced during dual fuel operation with gas injection tubes pointed down.

5.1.2 Fuel Consumption at Maximum Torque

Figure 5.2 shows the corresponding curves of brake specific fuel consumption (b.s.f.c.) as a function of engine speed in diesel only and dual fuel mode at maximum torque. In the calculations of the b.s.f.c. natural gas consumption was converted to energy equivalent of diesel fuel (using the factors discussed in Section 4.5.6). It is evident, from Figure 5.2, that the b.s.f.c. is higher in dual fuel modes than in straight diesel modes across the entire engine speed range. This is mainly due to natural gas losses with the scavenging air during dual fuel operation when part of the natural gas was pushed out of the cylinders before the exhaust valves were totally closed. The highest discrepancy between the b.s.f.c. in diesel only and dual fuel modes was at 800 rpm, and was found to diminish with increasing engine speed and decreasing the fraction of natural gas used. At 2200 rpm, where only a small fraction of the gaseous fuel was employed, the b.s.f.c. was at a similar level of about 290 g/kW.h for all test configurations.

The b.s.f.c. of the converted engine during diesel only operation was improved significantly compared with baseline diesel operation for engine speeds below 1600 rpm. This was mainly due to the fact that the converted engine was running leaner for this speed range. Above 1600 rpm, the b.s.f.c. stayed approximately at the same level for straight diesel operation prior and after conversion. The average b.s.f.c. of straight diesel operation across the engine speed range was 279.4 g/kW.h prior and 267.5 g/kW.h after

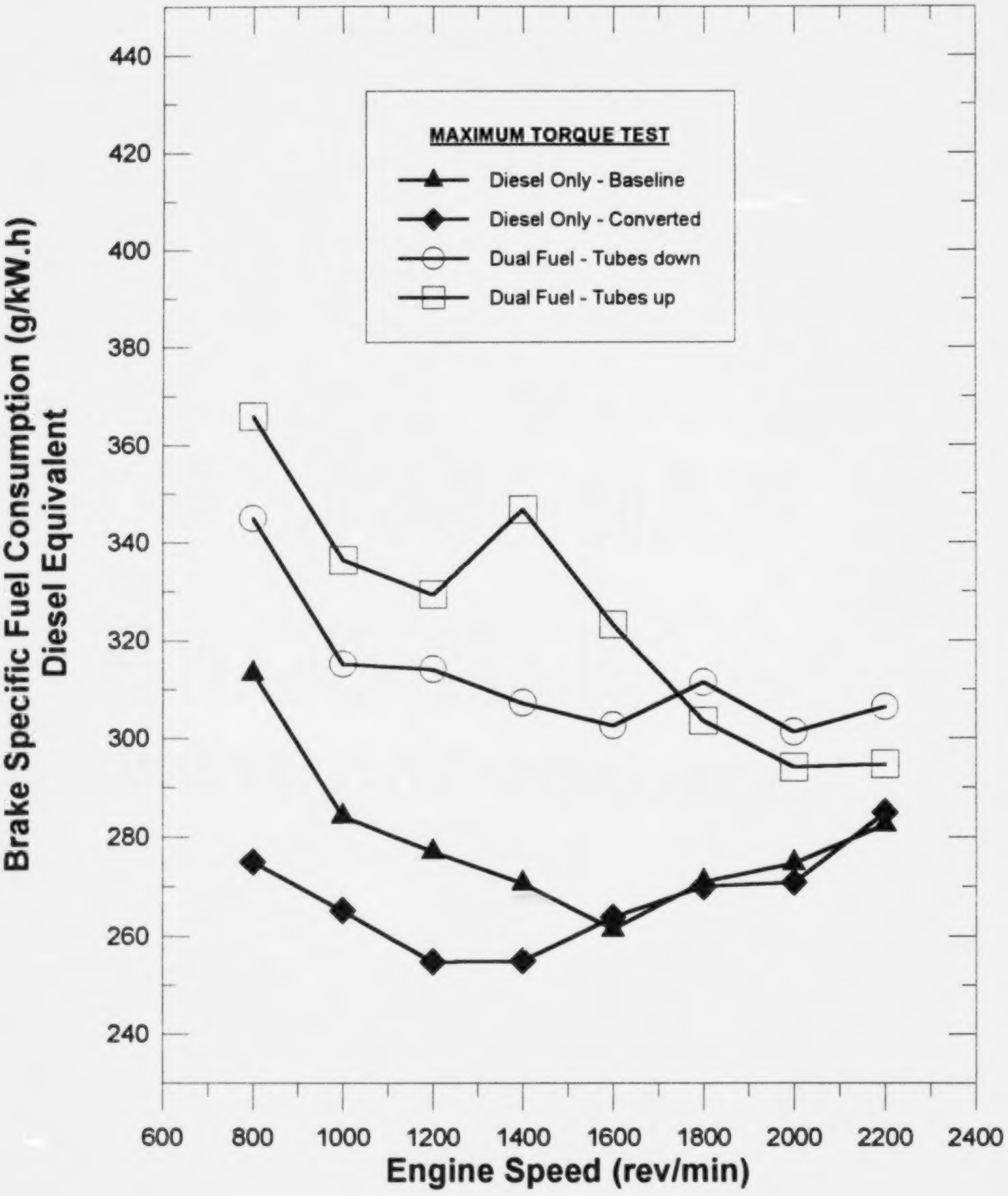


Figure 5.2 Brake specific fuel consumption in diesel only and dual fuel modes at maximum torque.

conversion. This resulted in 4.26 % reduction in the b.s.f.c. of the converted engine over the stock one during diesel only operation.

The total average b.s.f.c. (diesel fuel plus natural gas) during dual fuel operation was 312.9 g/kW.h and 324.2 g/kW.h for tubes down and tubes up configurations, respectively. Consequently, the b.s.f.c. increased 16.9 % for tubes pointed down and 21.2 % for tubes pointed up, when compared with diesel only operation of the converted engine.

5.1.3 Diesel Replacement at Maximum Torque

As defined earlier in chapter 4, diesel replacement ratio characterizes how much diesel fuel is replaced by natural gas in dual fuel operation compared with the amount of diesel fuel needed to produce the same load at the same engine speed in 100 % diesel operation. Figure 5.3 shows the diesel replacement obtained during dual fuel operation for both gas injection tubes configurations. Much more diesel fuel was replaced by natural gas at lower engine speeds. As the air intake temperature increased at higher speeds, more diesel pilot had to be used in order to avoid engine knock. The main factor influencing knock was a high air charge intake temperature, which was rising rapidly with engine speed as was presented earlier in Figure 4.6. Air charge temperatures as high as 115 °C were observed at 2200 rpm. Another factor which could contribute to the engine knocking problem was an elevated air box pressure (produced by the blower). According to the engine specifications, the air box pressure normally reaches 25.4 kPa above atmospheric at 2100 rpm and full load conditions. The engine's high compression ratio

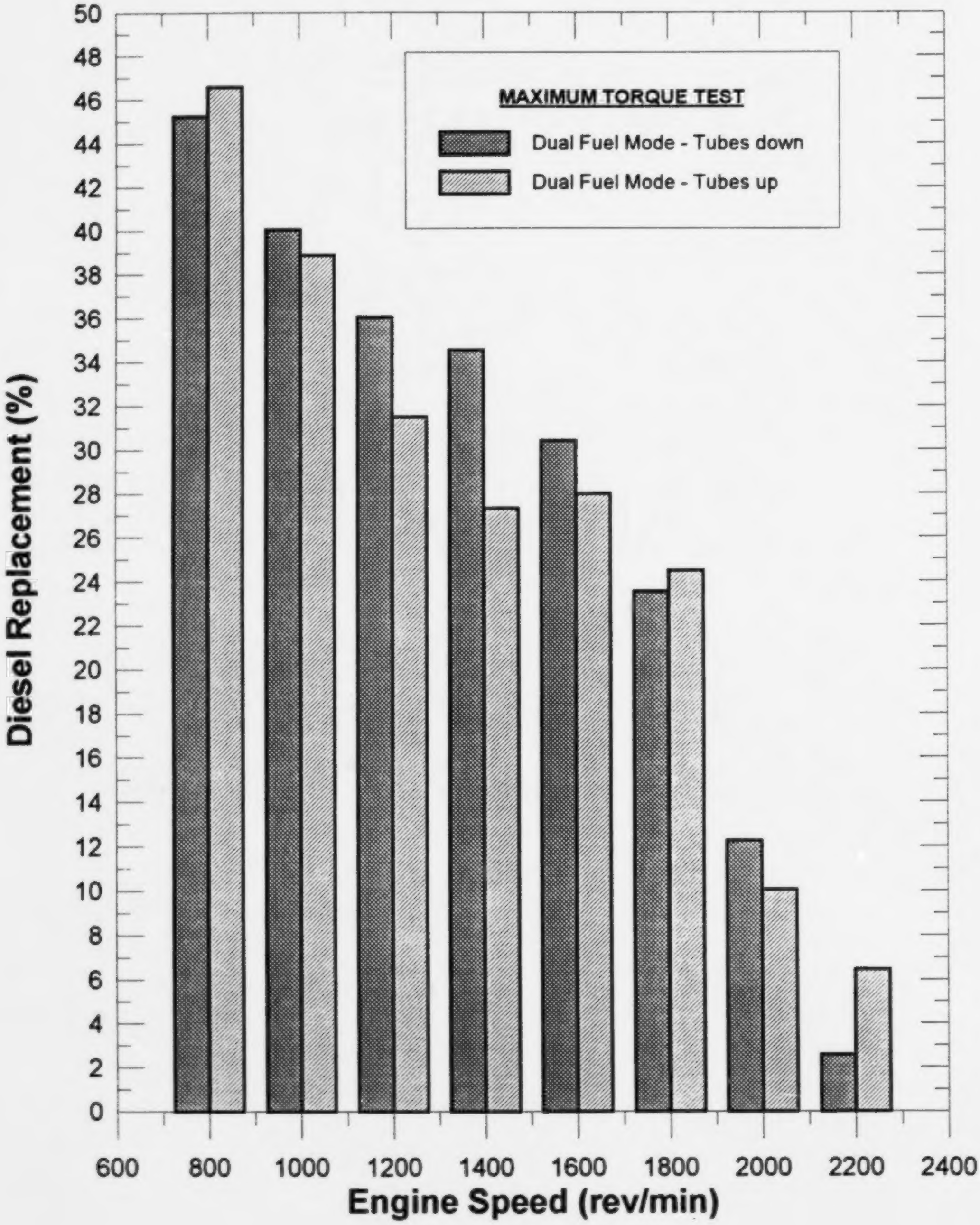


Figure 5.3 Diesel replacement ratio in dual fuel modes at maximum torque.

(CR=18.7:1), combined with hot intake conditions led to the onset of knock during dual fuel operation.

Consequently, the maximum diesel replacement ratio of 45.3 % and of 46.6 % was obtained at the same low engine speed of 800 rpm during dual fuel operation for tubes pointed down and tubes pointed up configurations, respectively. The diesel replacement ratio decreased with increasing engine speed, and reached the minimum level of 2.6 % for tubes down and of 6.5 % for tubes up configuration at 2200 rpm. The diesel replacement ratio averaged 28.1 % over the entire speed range for tubes pointed down configuration, (5 % higher than the average value of 26.7 % for tubes pointed up configuration).

5.1.4 Carbon Dioxide Emissions (CO_2) at Maximum Torque

Generally, carbon dioxide emissions are directly related to the engine power output and depend entirely on the amount and type of fuel burnt. The brake specific carbon dioxide emissions as a function of engine speed at maximum torque are presented in Figure 5.4. It is evident that the engine produced lower carbon dioxide emissions across the entire speed range when it was operated in dual fuel mode. However, a significantly higher reduction in CO_2 emissions was observed at lower engine speeds where more natural gas was utilized. For both diesel only and dual fuel operation, the carbon dioxide emissions increased monotonically with engine speed and reached their maximum values at 2200 rpm.

The average brake specific CO_2 emissions were 802 g/kW.h and 815 g/kW.h in

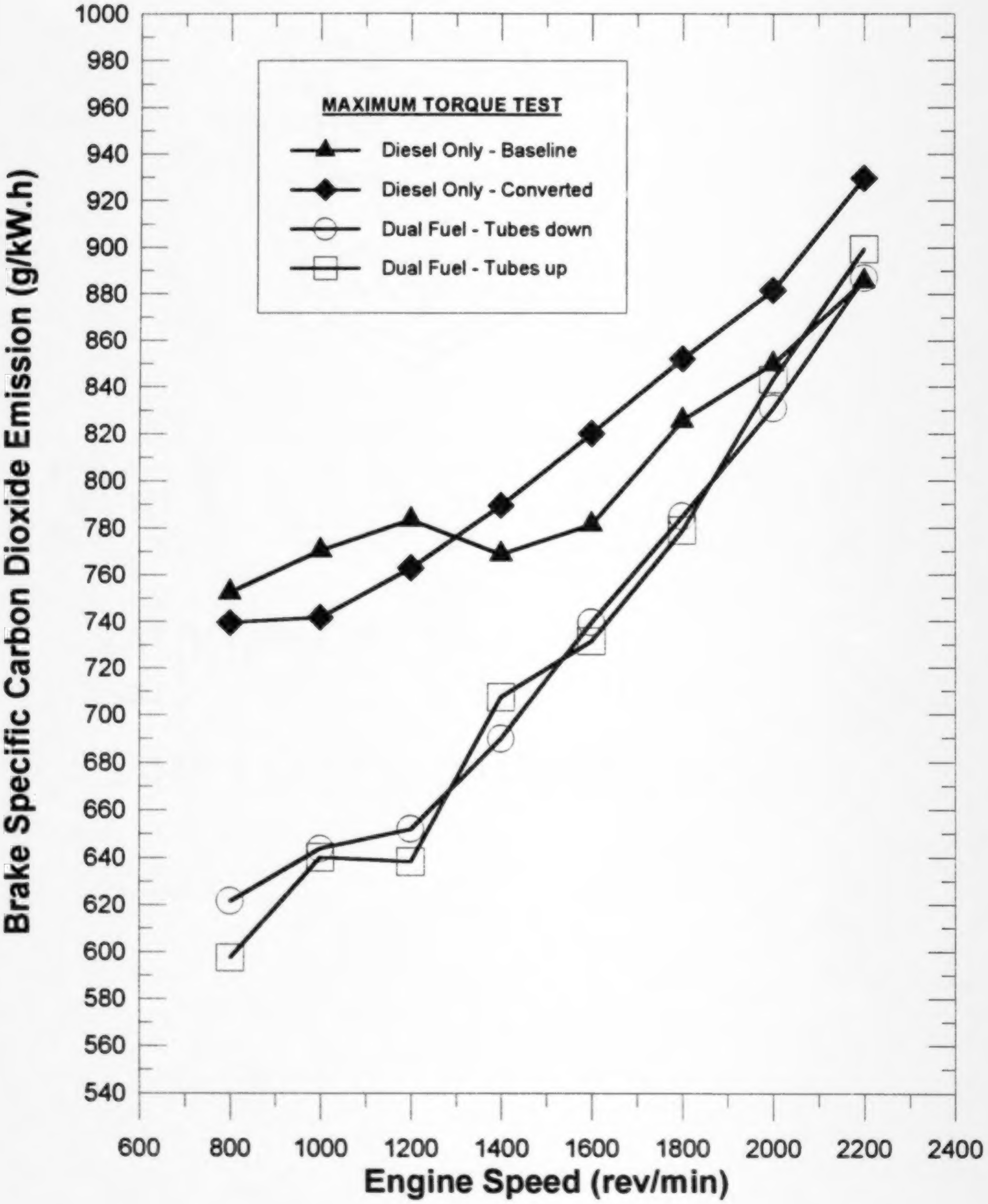


Figure 5.4 Brake specific carbon dioxide emissions at maximum torque.

diesel only mode before and after conversion, respectively. The configuration of the gas injection tubes had practically no effect on the CO_2 emissions during dual fuel operation. The average CO_2 emission levels were 731 g/kW.h and 730 g/kW.h for gas tubes deflected downwards and upwards, respectively. Hence, the average reduction in the CO_2 emissions in dual fuel mode was about 10.4 % when compared with straight diesel operation of the converted engine. The reduction in the CO_2 emissions results from higher hydrogen to carbon ratio (H/C) of natural gas, which directly impacts the number of CO_2 moles emitted. As the H/C ratio is increased, less carbon product, CO_2, is formed compared to hydrogen product, H_2O, thereby decreasing CO_2 emission level for dual fuel over straight diesel operation.

5.1.5 Carbon Monoxide Emissions (CO) at Maximum Torque

It is well known that carbon monoxide emissions result mainly from incomplete combustion of fuel. This pollutant is not normally a serious problem in diesel engines, which operate with an excess of air even at full load conditions. The diesel fuel injection system, which is responsible for in-cylinder mixing of diesel fuel with the surrounding air, plays an important role in CO formation. During dual fuel operation, the diesel pilot is injected into a preformed mixture of natural gas and air. When some regions of the premixed mixture are below the lower flammability limit, the gaseous fuel does not burn, but continues to undergo pre-flame oxidation reactions until it is exhausted from the cylinder. This can lead to increased production of CO emissions.

The brake specific carbon monoxide emission level at maximum torque, plotted

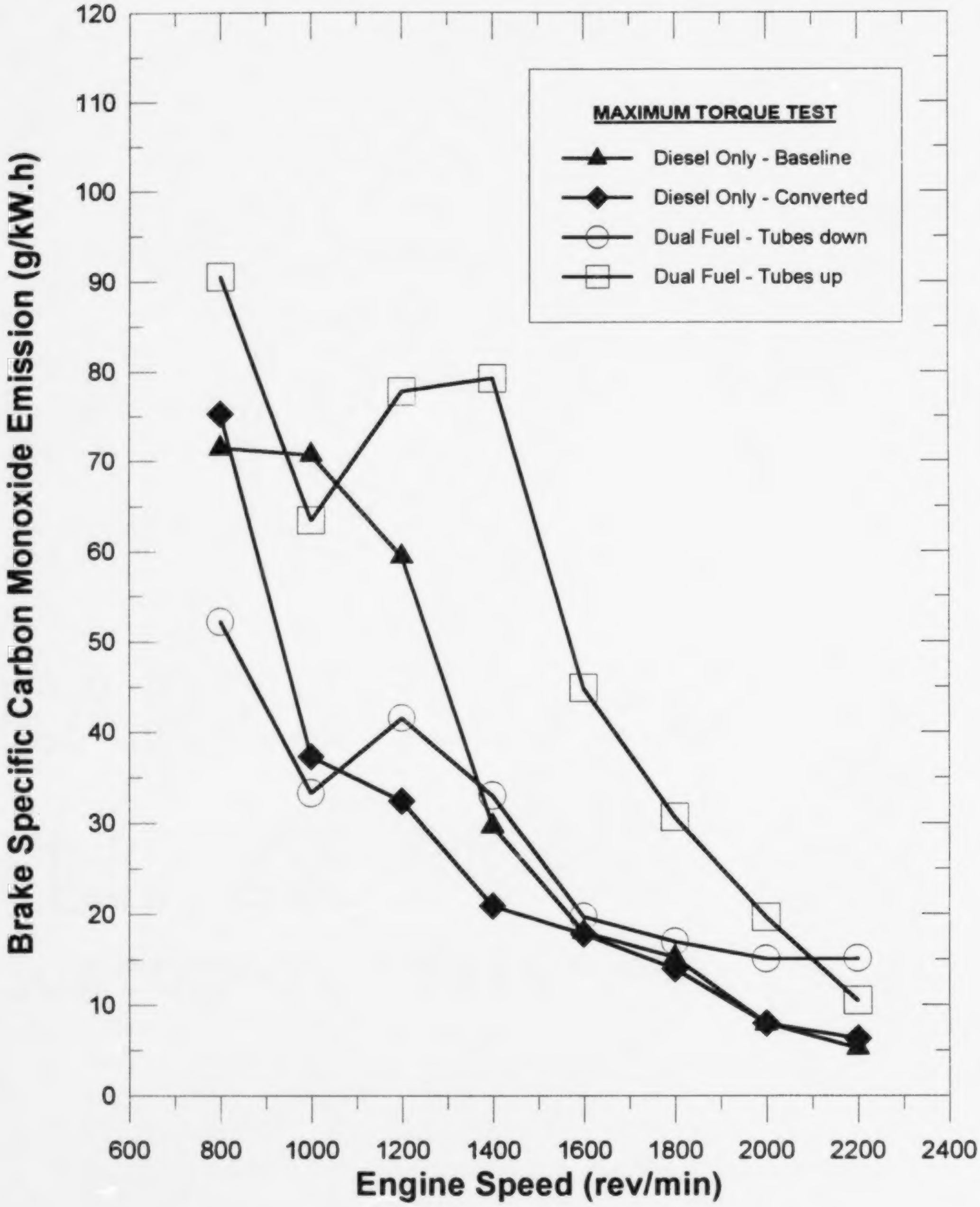


Figure 5.5 Brake specific carbon monoxide emissions at maximum torque.

against the engine speed, is presented in Figure 5.5. The CO curves follow a similar pattern for all operational modes: the CO emissions decreased substantially as the engine speed increased. The highest carbon monoxide emissions were observed at 800 rpm when the engine operated at the lowest overall air to fuel (A/F) ratios. In contrast, the lowest emissions were observed at 2200 rpm while the engine ran at the highest overall A/F ratios.

In diesel only mode, the average brake specific CO emissions were 34.7 g/kW.h and 26.5 g/kW.h before and after conversion, respectively. Particularly, the converted engine emitted significantly lower CO levels in the speed range between 1000 rpm and 1400 rpm. The lower corresponding fuel consumption levels and higher overall A/F ratios measured on the converted engine agree totally with the reduction in the CO emissions.

In dual fuel mode, the average CO emission levels were 28.3 g/kW.h and 52.1 g/kW.h for gas tubes deflected downwards and upwards, respectively. It is clear that the upward configuration of the gas injection tubes drastically raised the CO emissions during dual fuel operation. Consequently, the CO emissions averaged 96.7 % higher for the tubes pointed up configuration, when compared with straight diesel operation of the converted engine. When natural gas was injected through the tubes pointed down, the increase in the CO emissions was at the level of 7 %. The higher CO emissions produced during dual fuel operation are mainly due to both pre-flame oxidation reactions of the gaseous fuel and further oxidation of scavenged natural gas in the exhaust.

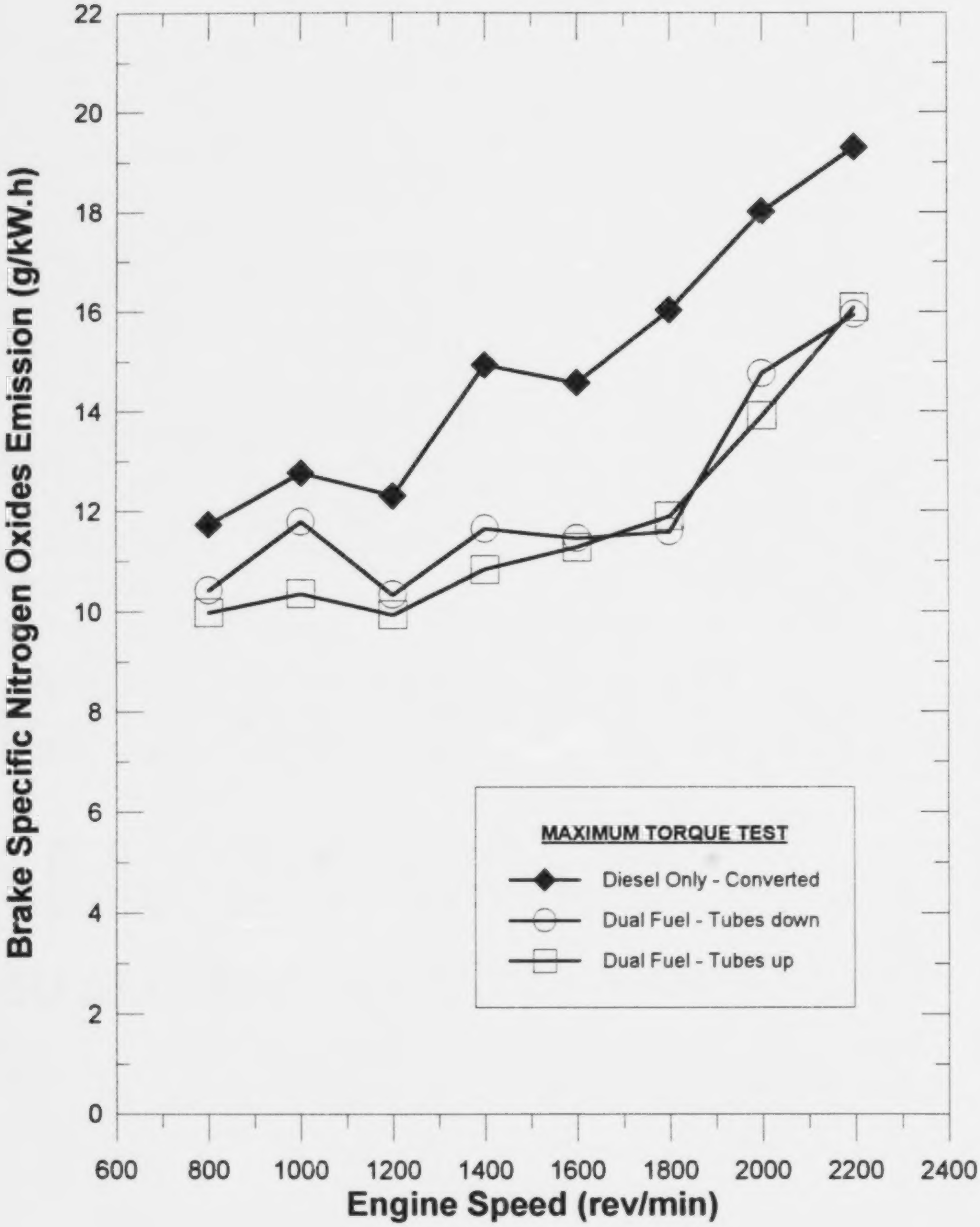


Figure 5.6 Brake specific nitrogen oxides emissions at maximum torque.

5.1.6 Oxides of Nitrogen Emissions (NO_x) at Maximum Torque

The brake specific oxides of nitrogen emissions increased almost linearly with engine speed, for both diesel only and dual fuel modes. This relationship is illustrated in Figure 5.6. The highest NO_x emissions were obtained during straight diesel operation for all engine speeds, averaging at the level of 14.9 g/kW.h. When the engine ran on dual fuel, the average NO_x emissions were 12.3 g/kW.h for tubes pointed down and 11.8 g/kW.h for the tubes pointed up configuration. Hence, there is no significant relationship between the type of injection tubes used and NO_x emissions at maximum torque conditions. For both tube configurations, dual fuel mode produced about 20 % less NO_x pollutant than diesel only mode. This reduction in NO_x emissions in dual fuel mode can be explained by the fact that a big part of the fuel burned under lean premixed conditions, with lower flame temperatures than the diesel diffusion flame. Also, overall combustion duration may be shorter with the partially premixed charge.

5.1.7 Reactive Non-Methane Hydrocarbons (NMHC) at Maximum Torque

Generally, the brake specific NMHC emissions, measured as equivalent hexane, were low for all test configurations across the entire engine speed range. The corresponding plots of NMHC emissions as functions of engine speed are presented in Figure 5.7. Because the non-methane hydrocarbons were calculated in dual fuel mode by subtracting two large values (total hydrocarbon and methane hydrocarbon values), there is some uncertainty in NMHC results. The highest average emissions (1.46 g/kW.h) were produced in dual fuel mode when tubes pointed upwards were

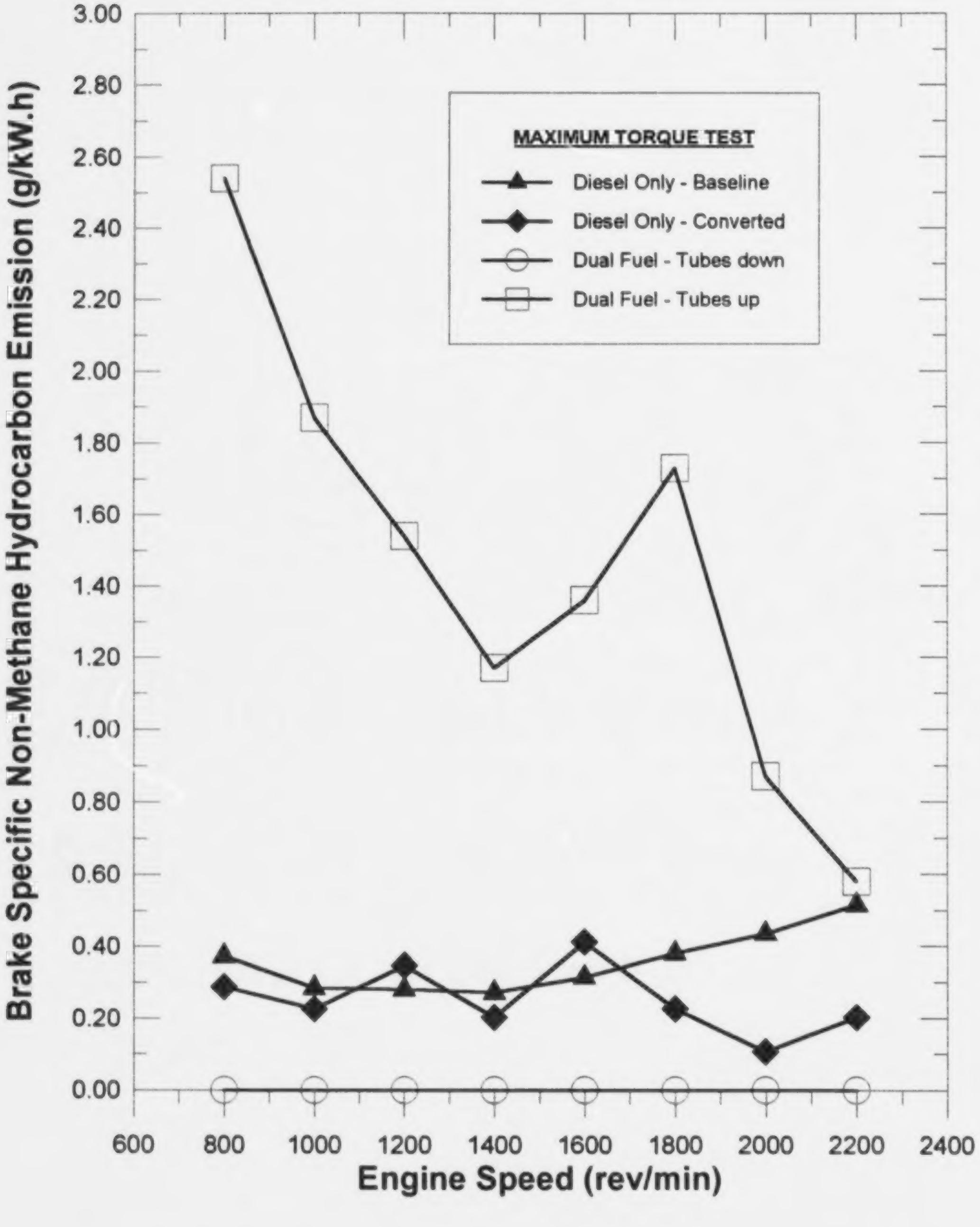


Figure 5.7 Brake specific non-methane hydrocarbon emissions at maximum torque.

employed. In contrast, there were no measurable NMHC emissions for the tubes pointed down configuration.

Diesel only operation averaged 0.36 g/kW.h and 0.25 g/kW.h before and after conversion, respectively. The emissions were practically independent of engine speed and were at a similar level for each case. The slightly lower production of NMHC emissions of the converted engine was probably due to lower diesel fuel consumption levels (higher A/F ratios) at maximum torque conditions. Consequently, better mixing and oxidation of diesel fuel with the surrounding air was achieved in the converted engine in diesel only mode.

5.1.8 Methane - Non Reactive Hydrocarbons (CH_4) at Maximum Torque

No significant methane fraction was found in the engine's exhaust in diesel only mode. In dual fuel mode, the brake specific methane emissions were found to decrease monotonically with increasing engine speed. The plots of the methane emissions as functions of engine speed for both dual fuel modes are shown in Figure 5.8. Apparently, methane emissions follow exactly the same relationship as diesel replacement ratio, illustrated earlier in Figure 5.3. In addition, a comparison of methane emissions with the excess specific fuel consumption in dual fuel mode (see Figure 5.9) shows that these two agree very closely. This confirms that natural gas scavenging losses were the primary source of the methane emissions.

The average methane emissions were 36.3 g/kW.h and 38.0 g/kW.h for tubes pointed down and up configuration, respectively. The former configuration performed

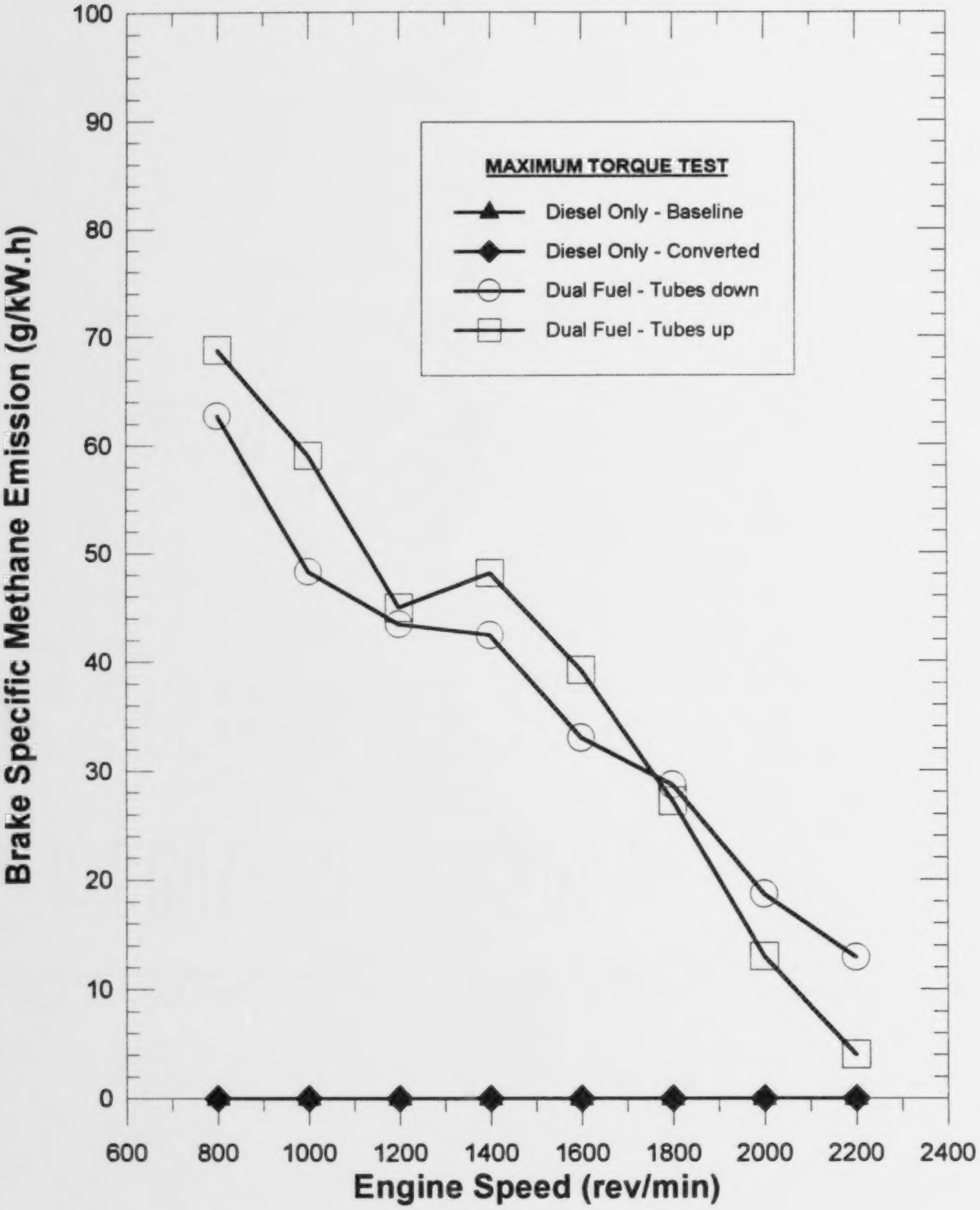


Figure 5.8 Brake specific methane emissions at maximum torque.

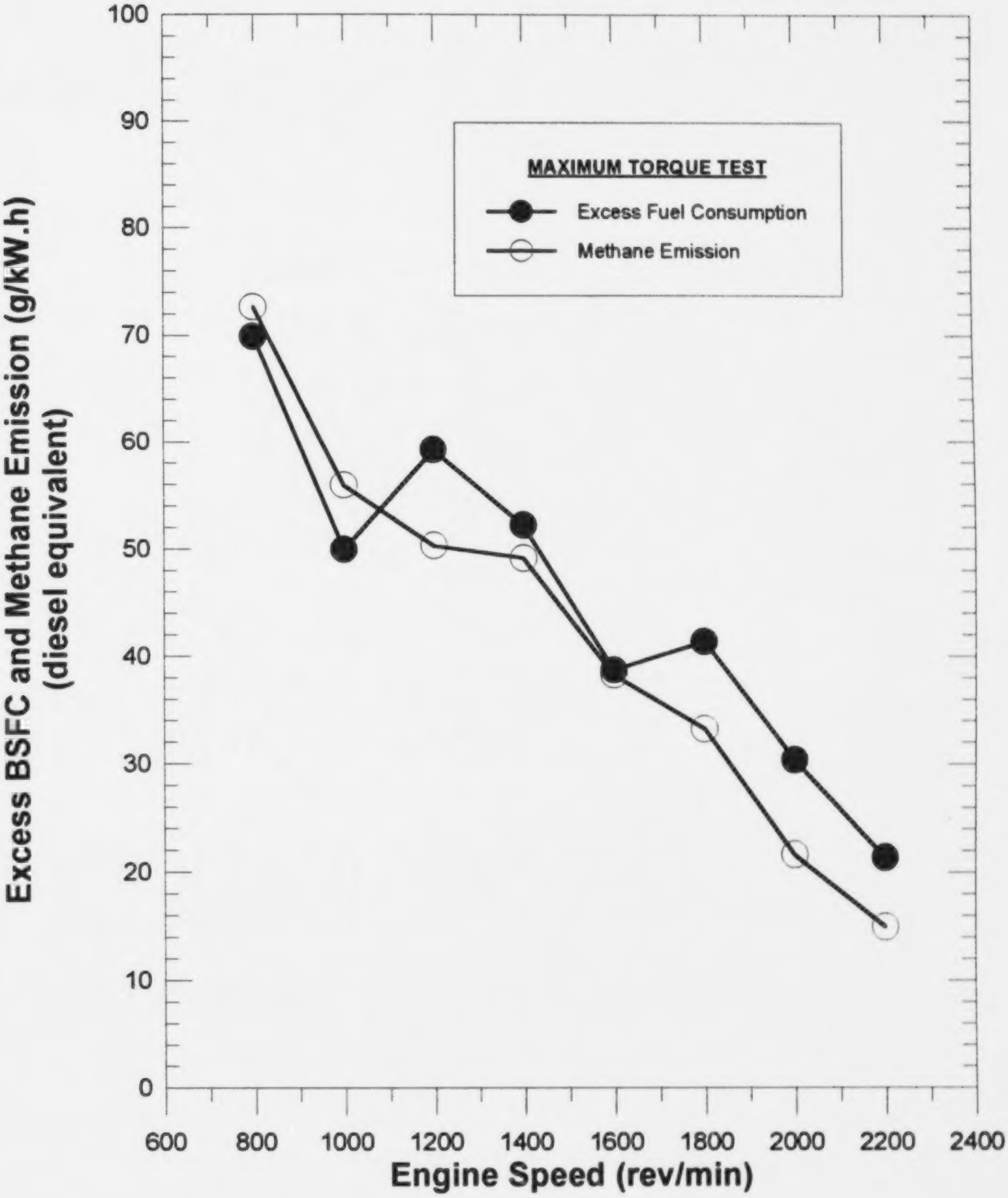


Figure 5.9 Comparison of methane emissions with the excess specific fuel consumption in dual fuel mode (tubes down) at maximum torque.

slightly better, producing about 5 % lower emissions than the latter one.

5.1.9 Soot Emissions at Maximum Torque

In pure diesel engines, soot formation from the engine cylinder is primarily dependent on the overall and the local air-fuel ratio. Because a significant portion of the diesel fuel/air mixture formation process happens during heterogenous combustion, local concentration of over-rich mixture occurs in the diffusion flame. This leads to an increase in soot production and emission of black smoke. In dual fuel engines, soot is formed mainly from the combustion of the diesel pilot injection. Many researchers have shown that dual fuel engines using high diesel fuel replacement ratios and small diesel pilots were capable of reducing soot formation [1, 2, 16]. However, an increase in soot emissions, (mainly due to longer ignition delay), is possible in engines using low diesel fuel replacement ratios and relatively large diesel pilots [2].

The brake specific soot emissions of the converted engine in diesel only and dual fuel modes are plotted in Figure 5.10. In general, the tested engine ran very clean in both modes emitting hardly any visible smoke even at the most severe conditions (low speed and maximum torque). In this study, a large reduction of soot was not expected because the maximum diesel fuel replacement ratio obtained was below 50 %, requiring use of large diesel pilots. As a matter of fact, the highest average brake specific soot emissions, 0.158 g/kW.h, were observed in dual fuel mode with injection tubes pointed up. The tubes pointed down configuration performed significantly better, giving the lowest soot emissions level of 0.062 g/kW.h. In diesel only mode, the soot emissions

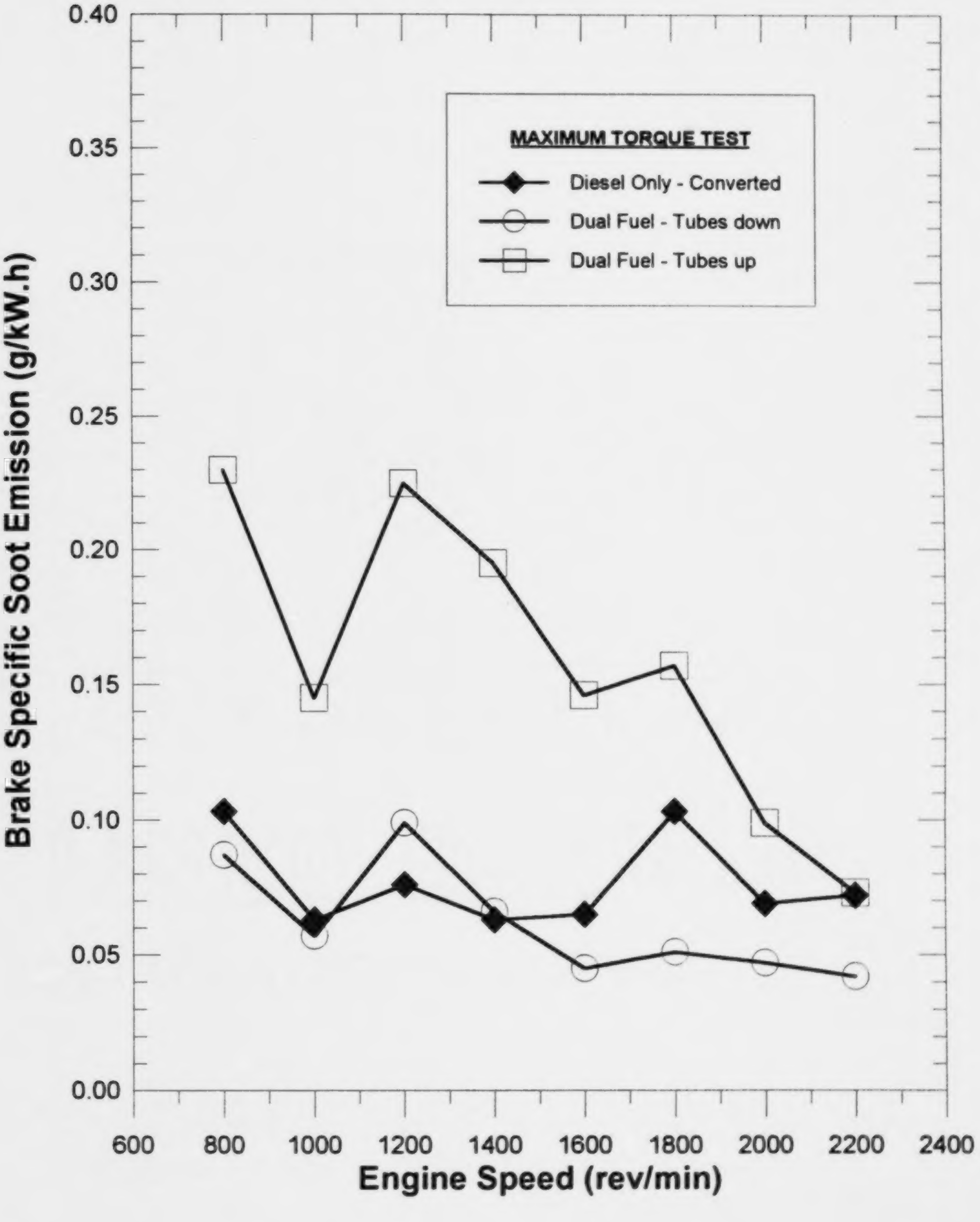


Figure 5.10 Brake specific soot emissions at maximum torque.

were slightly higher at the level of 0.077 g/kW.h. There was no clear relationship between the soot emissions and engine speed for the last two cases. However, in the case of dual fuel operation with tubes pointed up configuration, the soot emissions tended to decrease with increasing engine speed.

5.2 JAPANESE 13-MODE TEST RESULTS

The Japanese 13-mode test procedure was used to test the engine's emissions and fuel economy over a range of conditions typical for normal operation. The testing procedure emphasizes idling and medium torque operation rather than just maximum torque operation. Graphs of the brake specific fuel consumption and emissions plotted as a function of brake power for the multi-mode test are presented in this section. Modes 1 and 4, representing idle modes, and mode 13, representing a very low power mode, are operated on diesel fuel only and produce infinite or very high brake specific results. Therefore, they are not included in graphs illustrating the brake specific results. However, modes 1, 4, and 13 are included in the composite results calculations and these composite results for different test configurations are compared in this section.

5.2.1 Fuel Consumption

Figure 5.11 shows the corresponding plots of brake specific fuel consumption as a function of brake power in diesel only and dual fuel modes. In the calculations of the b.s.f.c., natural gas consumption was converted to energy equivalent of diesel fuel as described in Section 4.5.6. A comparison of the composite results is given in Figure 5.12. The composite fuel consumption results in dual fuel mode are shown as a sum of diesel fuel and natural gas fractions.

From Figure 5.11, it is evident that the b.s.f.c. was higher when the engine ran on dual fuel. In general, the b.s.f.c. decreased with increasing power for every test configuration tested. The higher power modes (80-100 kW) gave a b.s.f.c. of about

250 g/kW.h in diesel only modes, and of about 300 g/kW.h in dual fuel modes.

The composite fuel consumption in diesel only mode was 313 g/kW.h and 303 g/kW.h before and after conversion, respectively. This means that the converted engine reduced the composite fuel consumption by about 4.5 % when compared with baseline operation. In contrast, the total (diesel fuel and natural gas fraction) composite fuel consumption was significantly higher during dual fuel operation for both gas injection tubes configurations. However, the composite consumption of the diesel fuel fraction was reduced to 230 g/kW.h and to 234 g/kW.h for tubes pointed down and up, respectively. This corresponds to 24.1 % (73.1 g/kW.h) and 22.7 % (68.8 g/kW.h) reduction in the composite diesel fuel consumption. The composite natural gas consumption to provide these savings was 136 g/kW.h for tubes down and 137 g/kW.h for tubes up configuration. Only 51.2 % (tubes down) and 48.0 % (tubes up) of supplied natural gas was used effectively to replace diesel fuel. The rest was apparently scavenged directly to the exhaust line. Consequently, the total energy consumption increased about 23 % and 26 % in dual fuel modes with tubes deflected down and tubes deflected up, respectively.

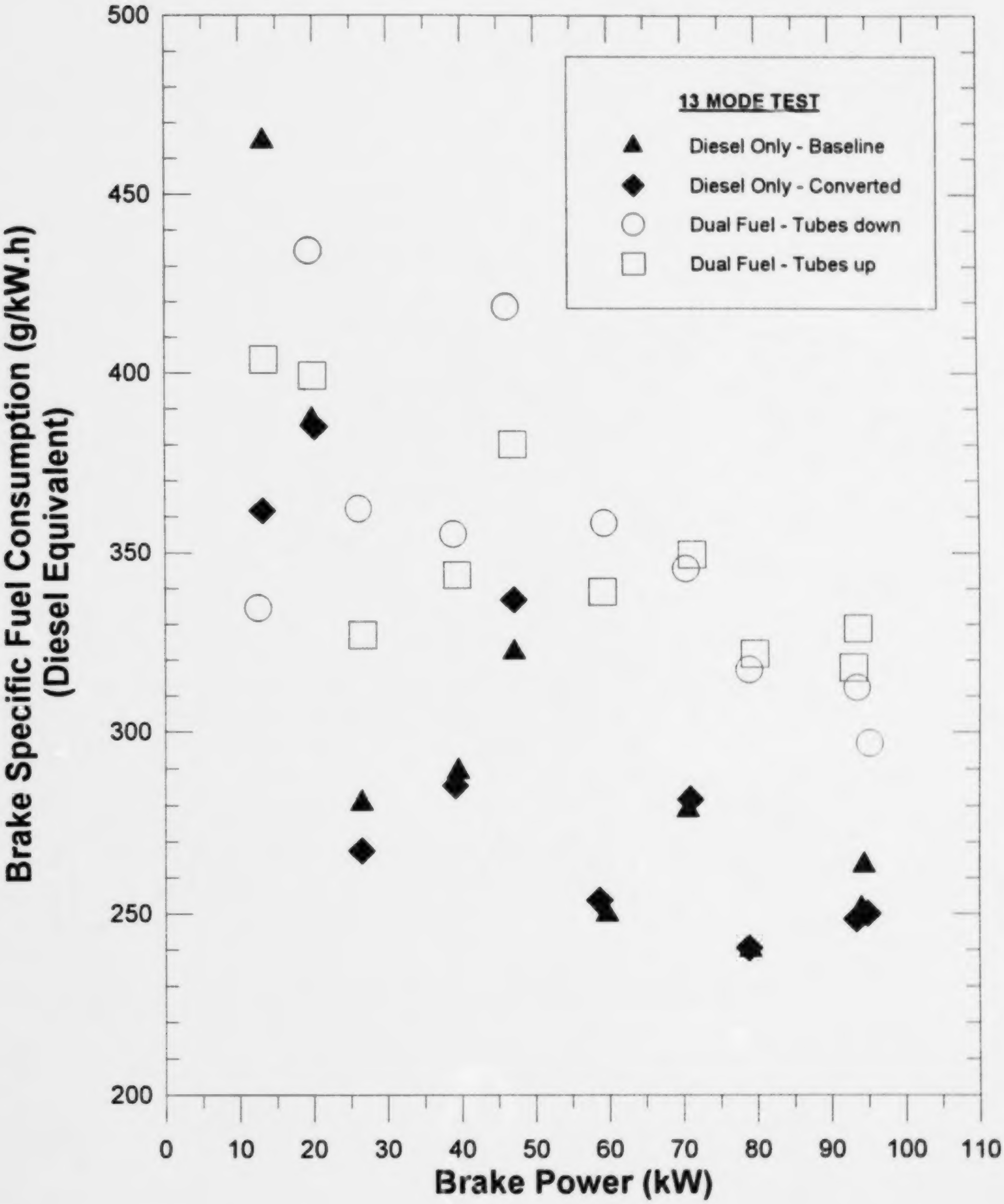


Figure 5.11 Brake specific fuel consumption in diesel only and dual fuel modes at part load (13-mode test).

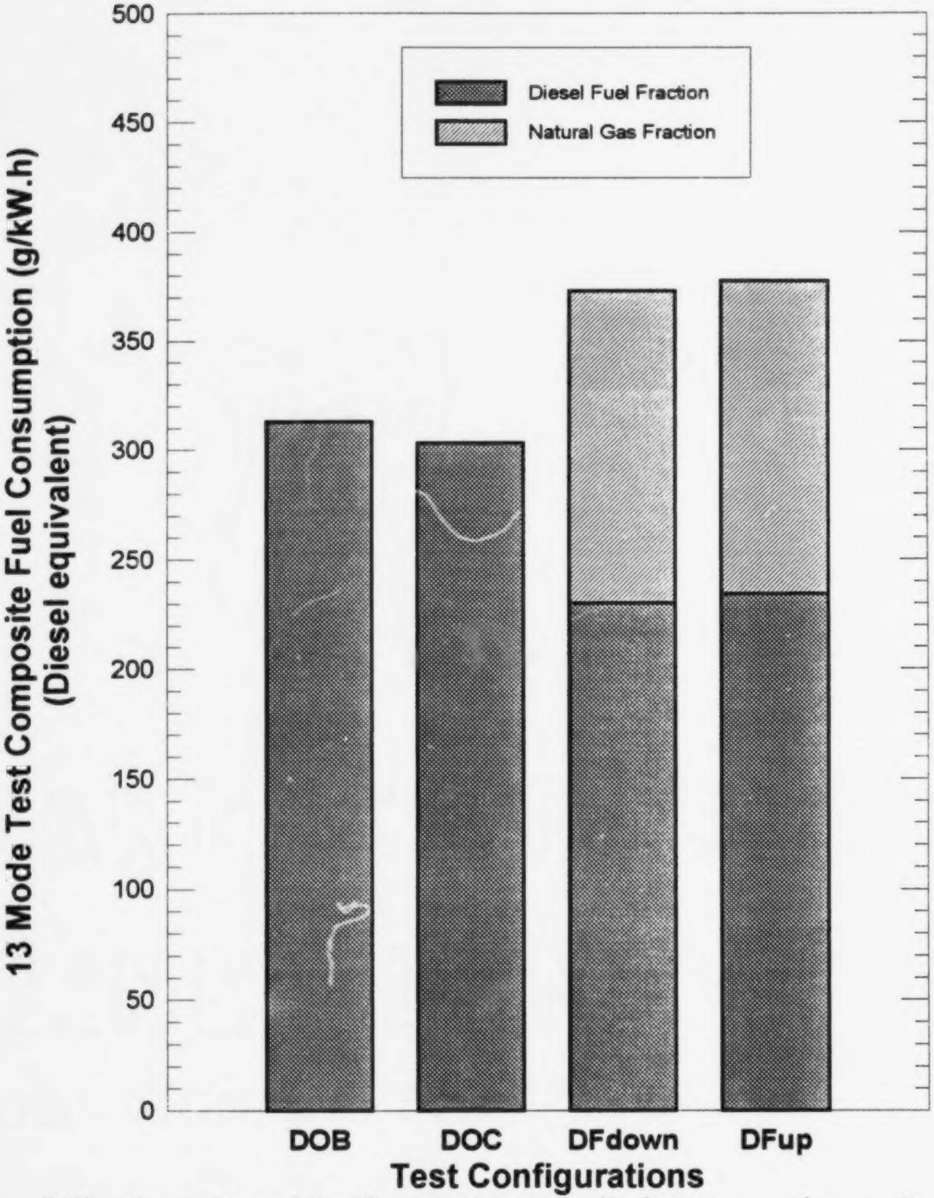


Figure 5.12 Comparison of the 13-mode test composite fuel consumption results.

Where:
DOB - Diesel Only-Baseline
DOC - Diesel Only-Converted
DFdown - Dual Fuel-Tubes down
DFup - Dual Fuel-Tubes up

5.2.2 Diesel Replacement

Figure 5.13 shows the diesel fuel replacement ratio, obtained during multi-mode test, as a function of brake power. The lowest diesel replacement levels were observed at low power modes, where the total fuel consumption was low and relatively high diesel pilots had to be used to ignite very lean mixtures of air and natural gas. Increasing the diesel pilot increases the volume of the charge affected by the combustion of the pilot fuel, and thus increases the fraction of natural gas that burns and decreases the pollutants emitted in the exhaust. As mentioned earlier, no natural gas was used during modes 1 and 4, at engine idle, and during mode 13, which represents only 5 % of full load at 1320 engine rpm. When engine load increased, relatively lower diesel pilots could be used and higher diesel replacement ratios were obtained. The maximum diesel replacement was measured in mode 9 (about 60 kW at 1320 engine rpm) for both gas injection tube configurations. The diesel replacement ratio was 34.3 % with tubes pointed down, and 38.2 % with tubes pointed up. As the brake power exceeded a level of about 70 kW, the air charge temperature increased significantly leading to the onset of engine knock. Relatively higher diesel pilots and smaller natural gas fractions were used to avoid it. Therefore, slightly lower diesel substitution rates were measured in modes 10, 11, and 12.

A comparison of the composite diesel replacement results for both dual fuel testing configurations are presented in Figure 5.14. Tubes pointed down performed better and achieved the composite diesel replacement ratio of 24.1 %. The corresponding result for the tubes pointed up configuration was 5.8 % lower and was at a level of 22.7 %.

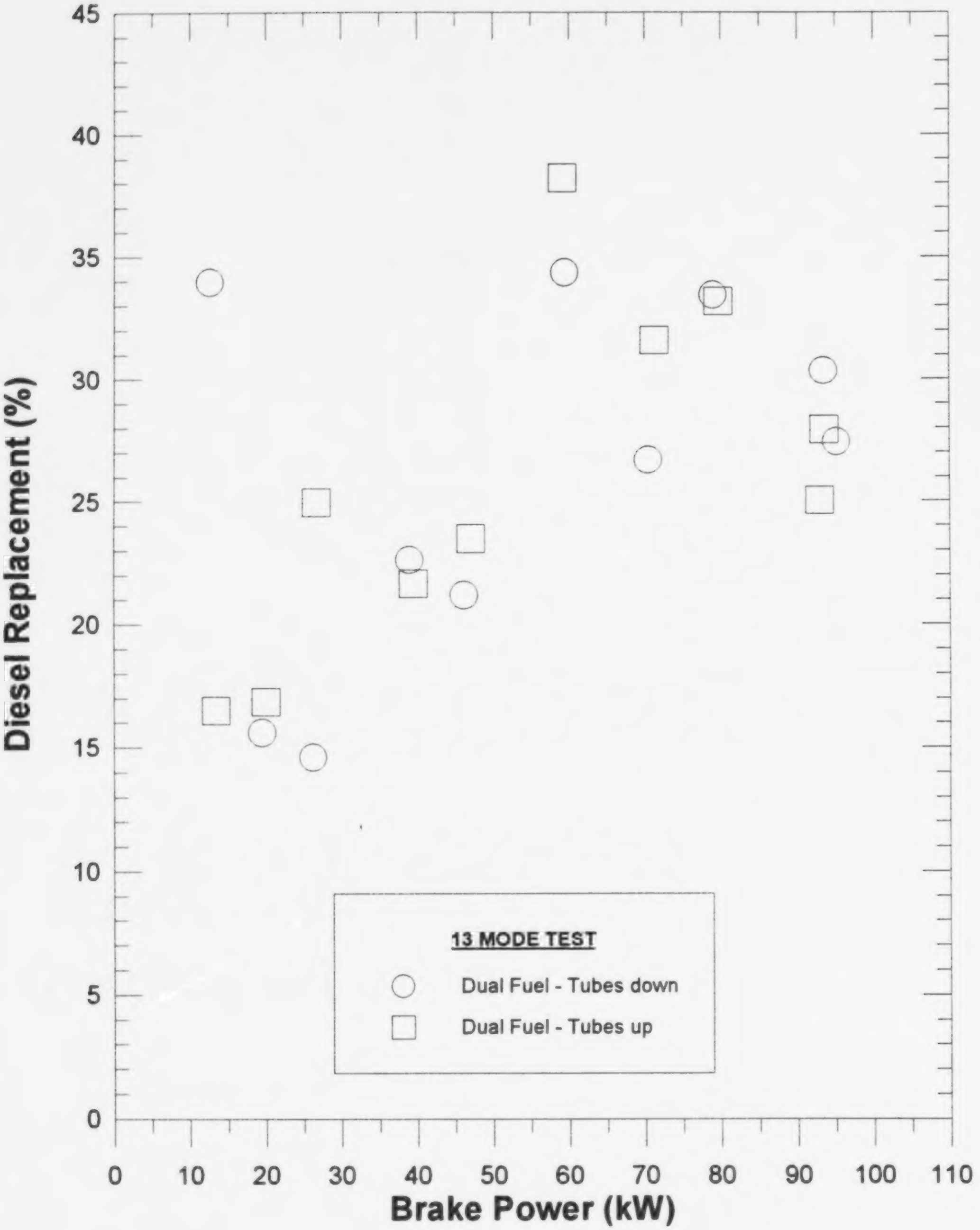


Figure 5.13 Diesel replacement ratio in dual fuel modes at part load (13-mode test).

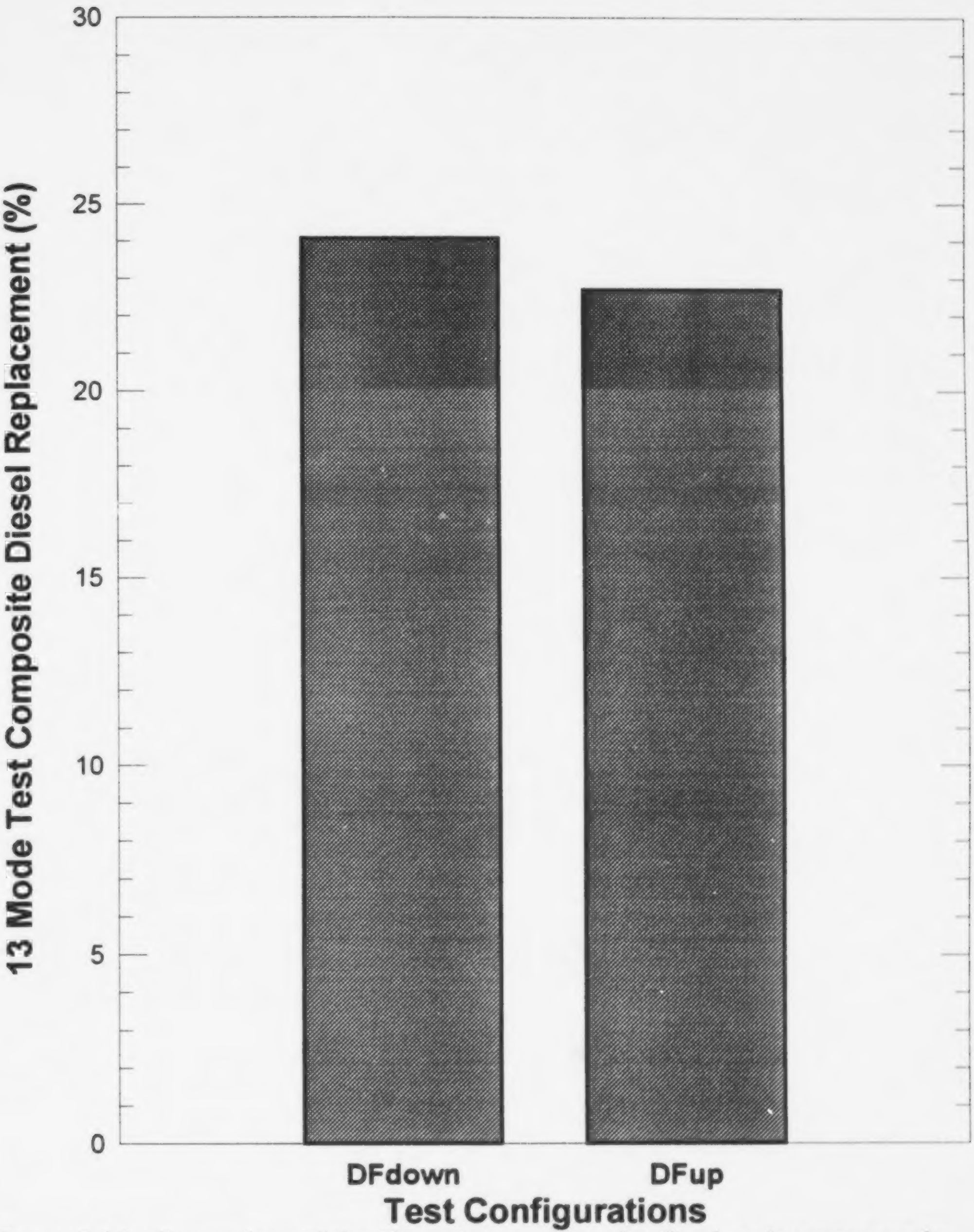


Figure 5.14 Comparison of the 13-mode test composite diesel replacement results.

Where:
DFdown - Dual Fuel-Tubes down
DFup - Dual Fuel-Tubes up

5.2.3 Carbon Dioxide Emissions (CO_2)

The CO_2 emissions for all test configurations as a function of brake power are presented in Figure 5.15. Generally, the brake specific carbon dioxide emissions had a very similar relationship with power as the brake specific fuel consumption. Like the b.s.f.c., the brake specific CO_2 emissions decreased with increasing power. From about 1300 g/kW.h for the low power modes, the CO_2 emissions decreased monotonically to about 700 g/kW.h for the higher power modes. As expected, dual fuel operation produced lower CO_2 emissions than diesel only operation for all testing points. The reduction in CO_2 emissions was primarily due to higher H/C (hydrogen to carbon) ratio of natural gas fuel fraction.

A comparison of the composite CO_2 emissions is illustrated in Figure 5.16. Diesel only operation of the baseline engine produced the highest composite CO_2 emissions, which were at the level of 963 g/kW.h. The converted engine emitted 948 g/kW.h when run in diesel only mode. The reduction in CO_2 emissions is related directly to the lower composite diesel fuel consumption of the converted engine in diesel only mode. In dual fuel mode, the composite CO_2 emissions were further reduced for both gas tubes configurations. The composite results were 890 g/kW.h for tubes pointed down, and 919 g/kW.h for tubes pointed up configuration. This corresponds to 6.2 % and to 3.0 % reduction when compared with straight diesel operation of the converted engine for tubes deflected down and tubes deflected up, respectively.

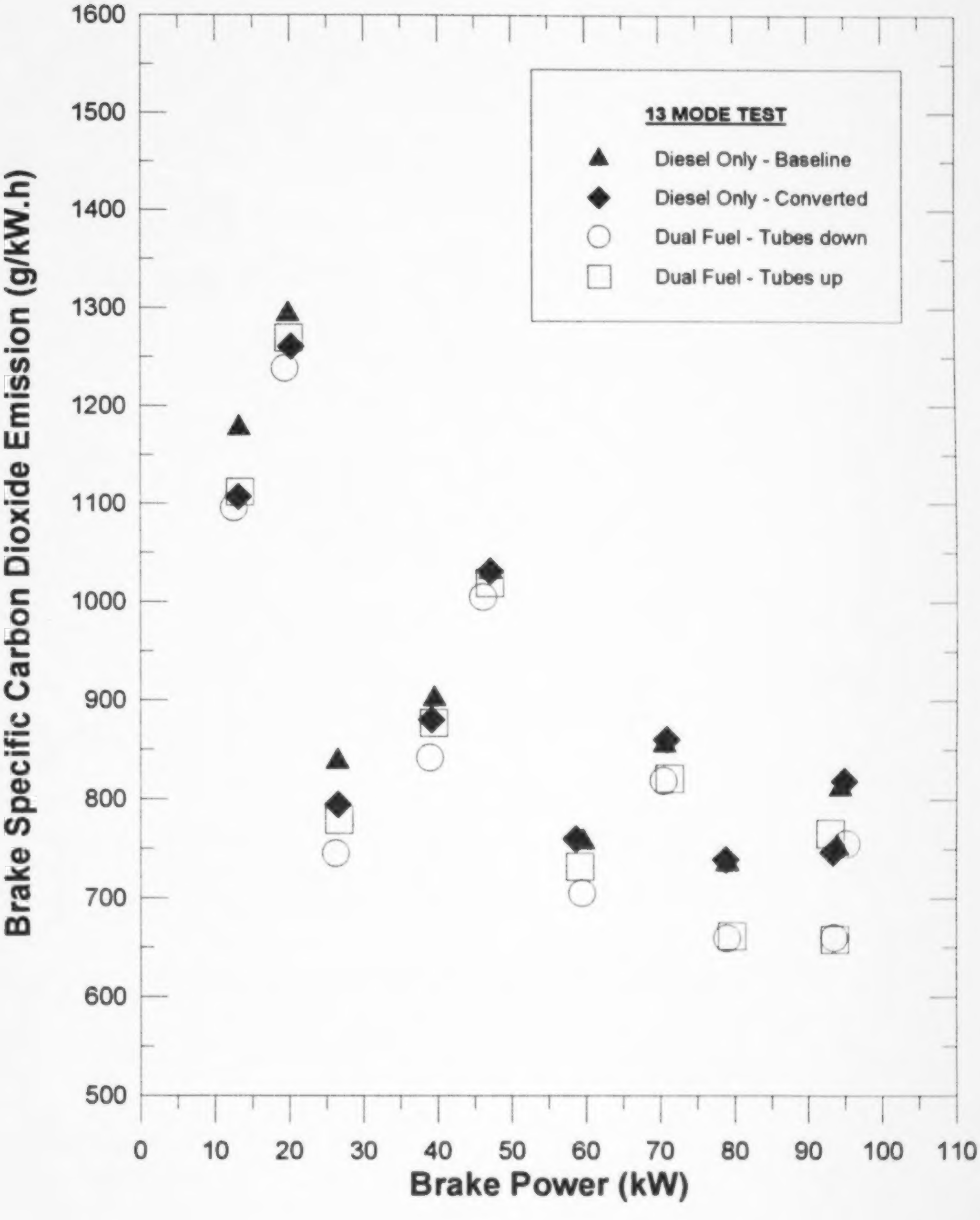


Figure 5.15 Brake specific carbon dioxide emissions in diesel only and dual fuel modes at part load (13-mode test).

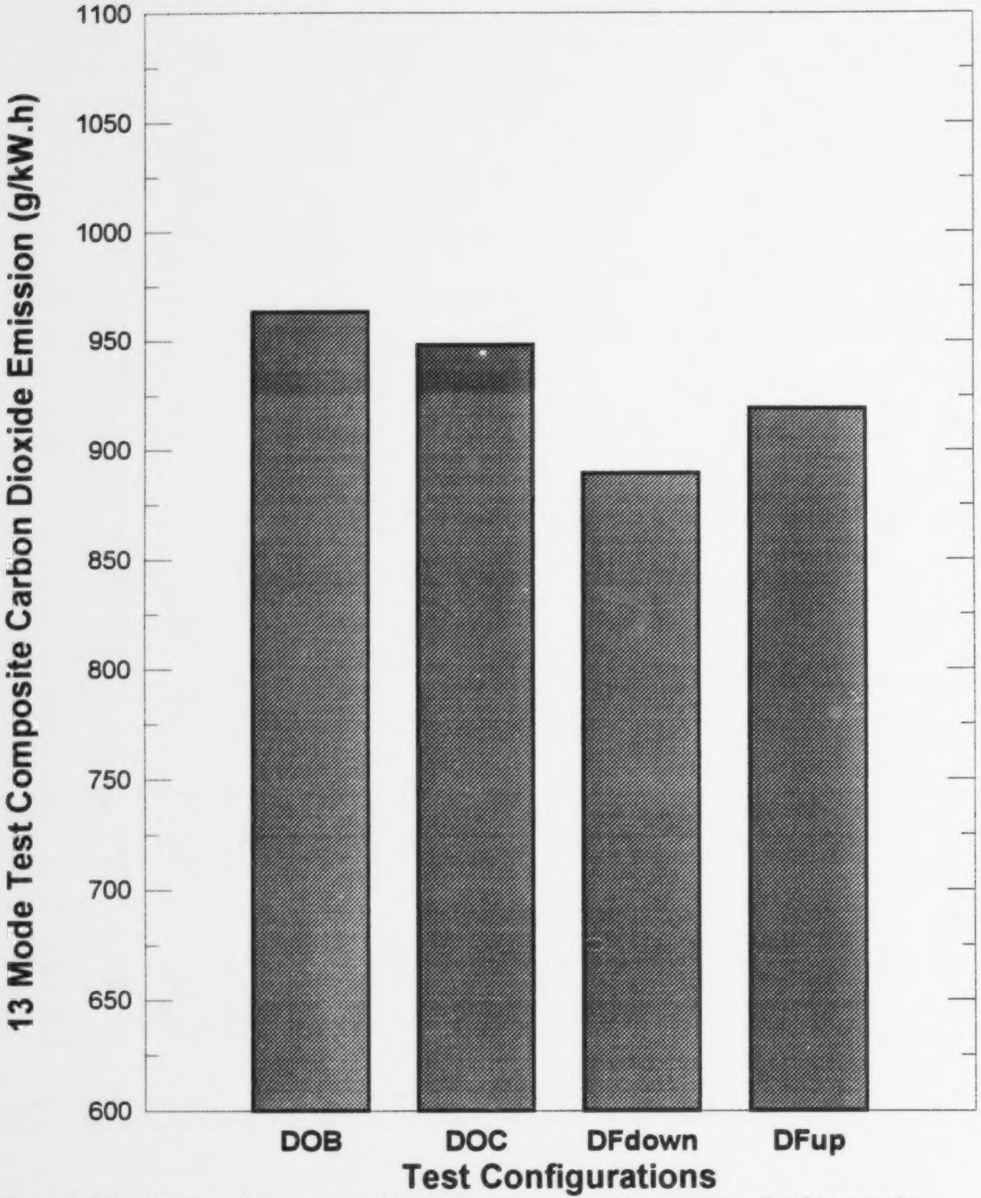


Figure 5.16 Comparison of the 13-mode test composite carbon dioxide results.

Where:
DOB - Diesel Only-Baseline
DOC - Diesel Only-Converted
DFdown - Dual Fuel-Tubes down
DFup - Dual Fuel-Tubes up

5.2.4 Carbon Monoxide Emissions (CO)

The brake specific carbon monoxide emissions as a function of brake power in diesel only and dual fuel modes are shown in Figure 5.17. In general, very low levels of CO emissions were measured when the engine was operated on diesel fuel only. The converted engine produced slightly lower CO emissions across the entire power range than the stock engine. This reduction in CO emissions agrees totally with the slightly higher overall A/F ratios observed on the converted engine during straight diesel operation.

It is evident, from Figure 5.17, that dual fuel operation produced significantly higher brake specific CO emissions for all testing points. The CO plots for both dual fuel modes follow a similar pattern: the CO emissions were relatively high at low power modes (less than 25 kW), very low at moderate power modes (25-70 kW), and maximum at high power modes (above 70 kW). The major source of high CO emissions under low load conditions was the occurrence of air fuel mixtures in the cylinder that are too lean to burn efficiently. This resulted in emission of partially reacted fuel in the exhaust. At moderate power loads, significantly richer mixtures of natural gas with air were employed and the combustion process was more complete. As a result, the best CO emission levels were obtained at moderate load conditions; however, they were still higher than the corresponding diesel-only levels. At high load and high engine speed conditions (modes 11, and 12), the natural gas fuel fraction was decreased while the diesel pilot was enlarged to avoid the risk of knocking combustion. Consequently, the mixtures of natural gas with air were leaner again, and large amounts of partial reaction

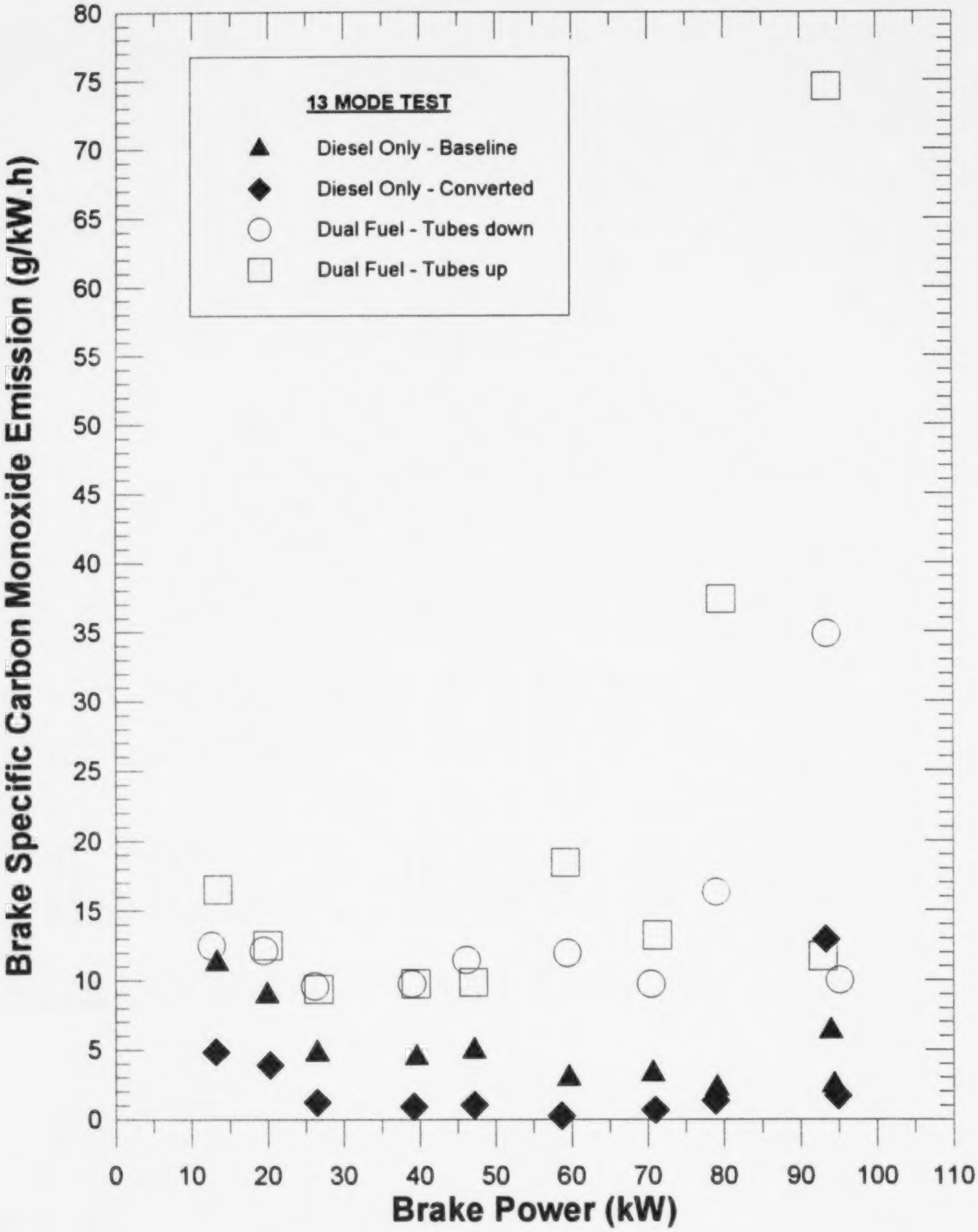


Figure 5.17 Brake specific carbon monoxide emissions in diesel only and dual fuel modes at part load (13-mode test).

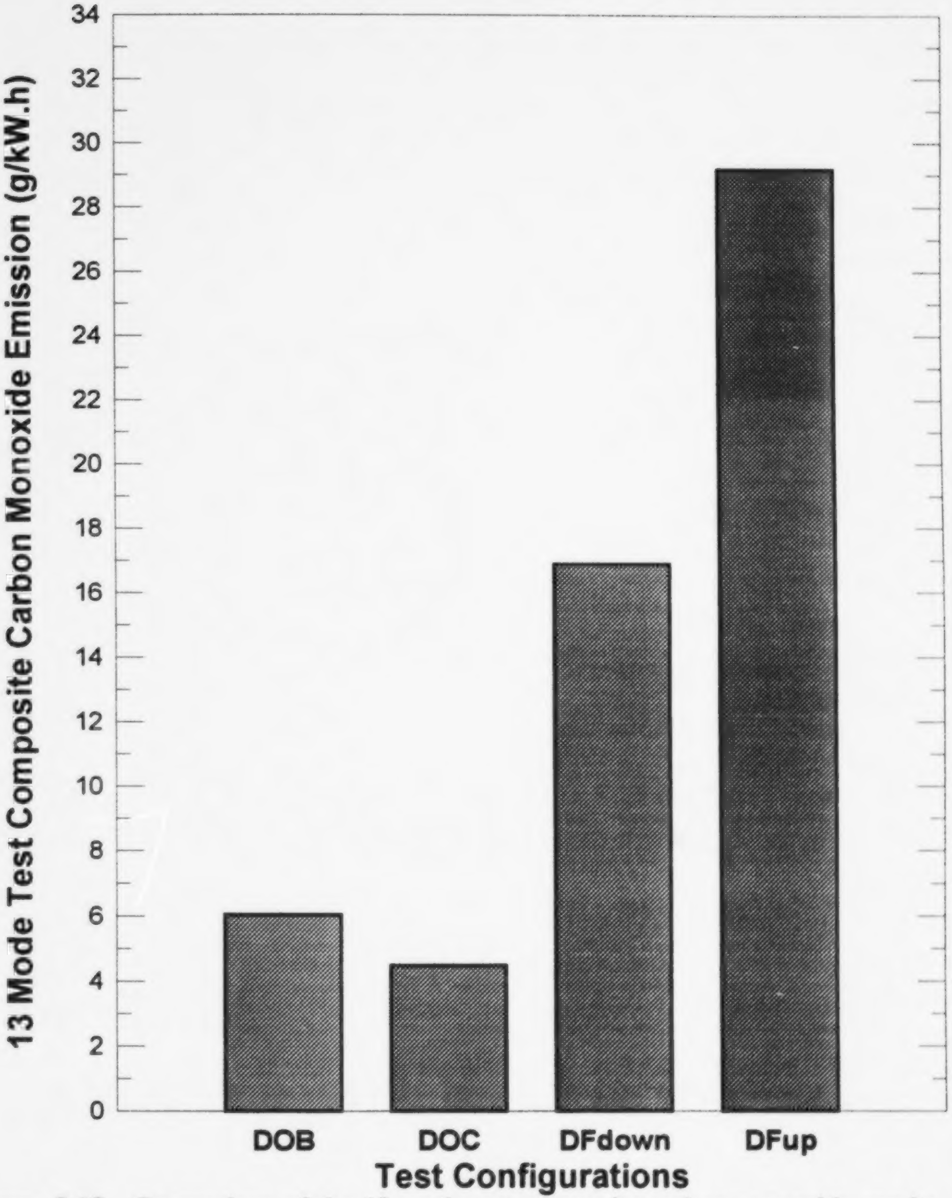


Figure 5.18 Comparison of the 13-mode test composite carbon monoxide results.

Where:
DOB - Diesel Only-Baseline
DOC - Diesel Only-Converted
DFdown - Dual Fuel-Tubes down
DFup - Dual Fuel-Tubes up

products were emitted.

The composite CO emissions for all test configurations are shown in Figure 5.18. For diesel only operation, the composite CO emissions were 6.06 g/kW.h before conversion and 4.48 g/kW.h after conversion. This corresponds to 26 % reduction in CO emissions of the converted engine over the stock one. On dual fuel, the CO emissions were about 4-6 times higher compared to diesel only operation of the converted engine. The composite results were 16.9 g/kW.h and 29.2 g/kW.h for tubes pointed down and for tubes pointed up configurations, respectively. These results show that tubes pointed down performed better, producing only about half of the CO emissions obtained with tubes pointed up.

5.2.5 Oxides of Nitrogen Emissions (NO_x)

The brake specific nitrogen oxides emissions as a function of brake power for diesel only and dual fuel operation are presented in Figure 5.19. Dual fuel operation reduced the NO_x emissions over the entire brake power range compared to straight diesel operation. The reduction in NO_x emissions is primarily due to the fact that the natural gas fuel fraction burnt under lean premixed conditions, where the combustion temperatures were lower than in a straight diesel diffusion flame. The brake specific NO_x emissions did not change with brake power and were almost on the same level for all testing points (modes) for each particular test configuration. The absolute emissions levels, however, increased almost linearly with increasing power in both diesel only and dual fuel modes.

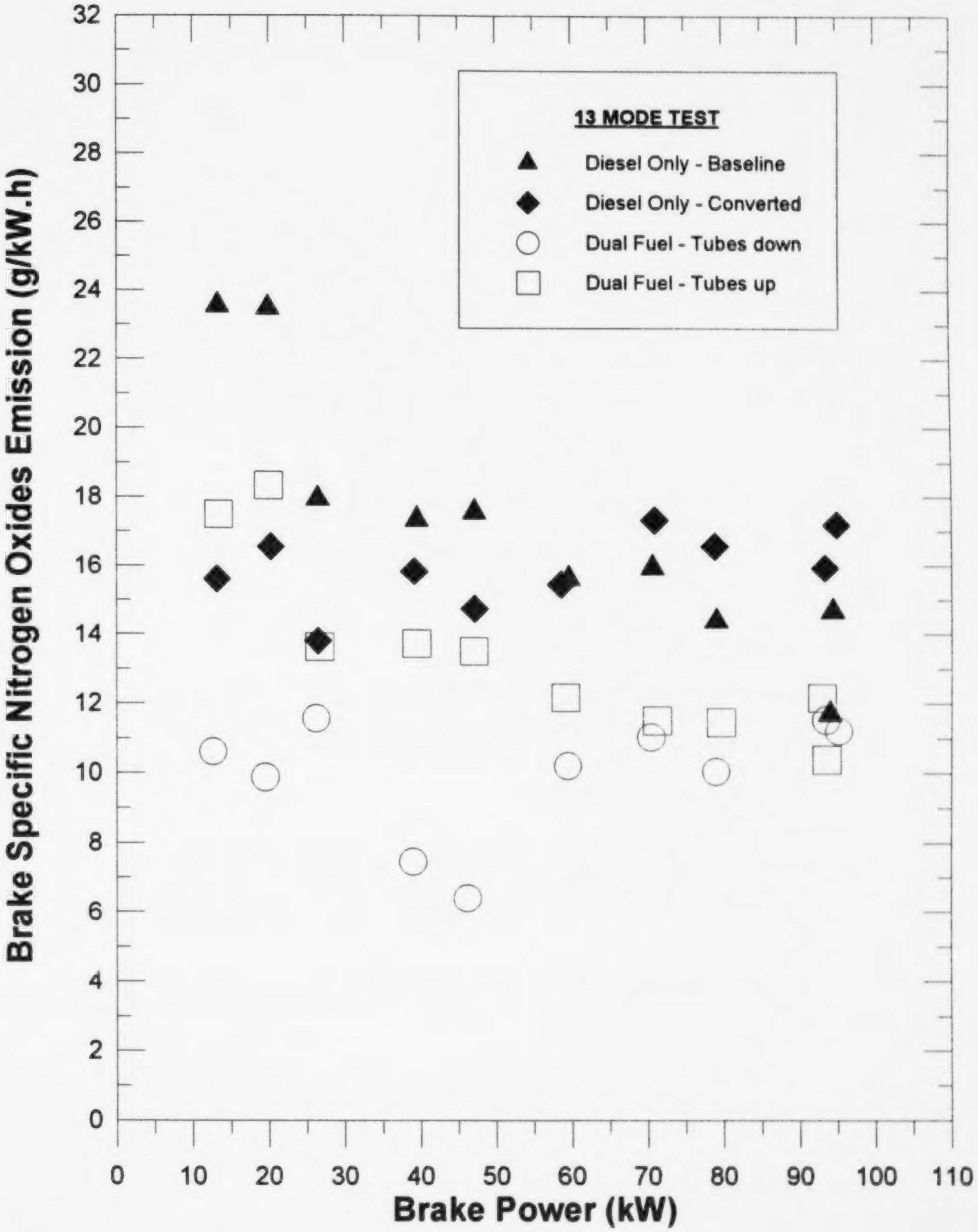


Figure 5.19 Brake specific oxides of nitrogen emissions in diesel only and dual fuel modes at part load (13-mode test).

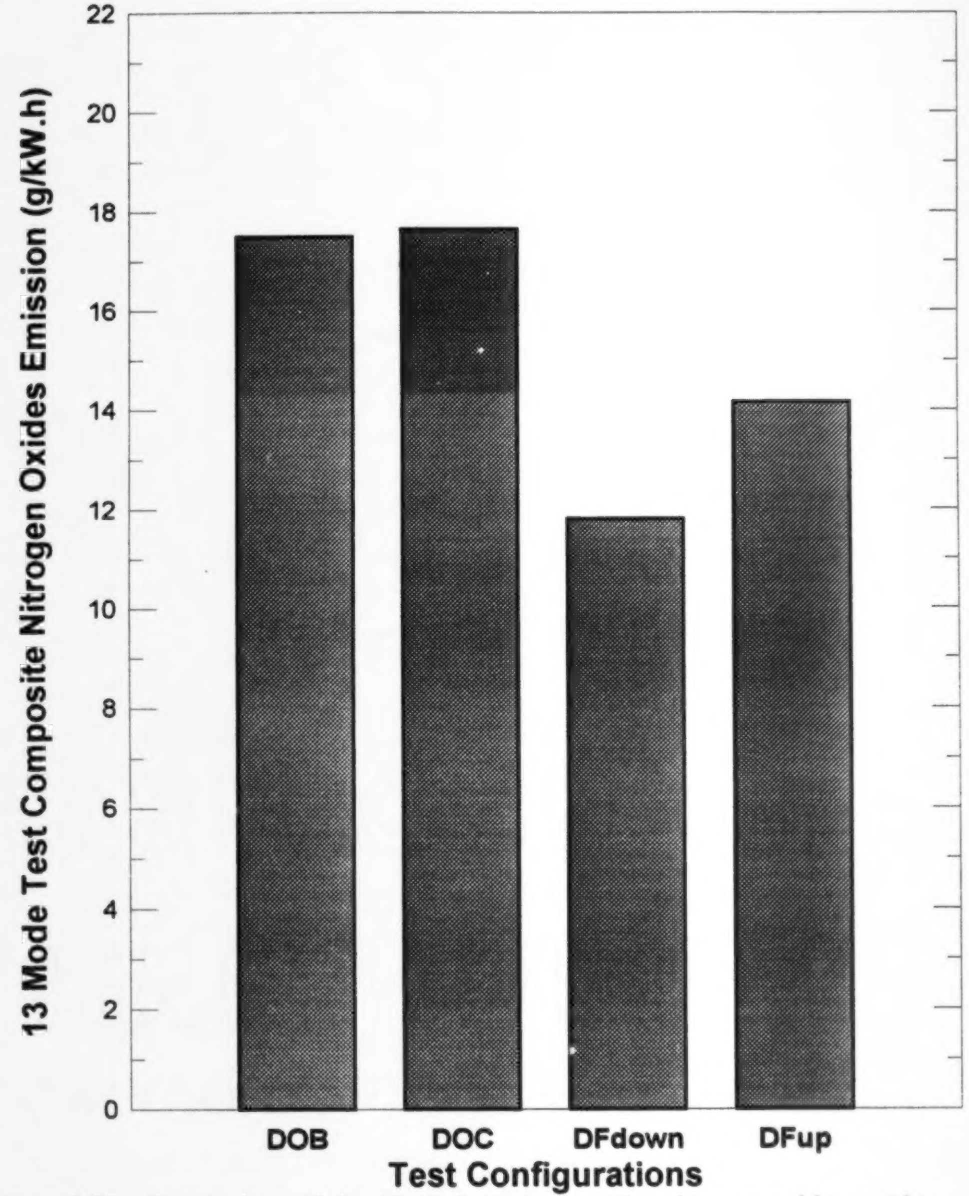


Figure 5.20 Comparison of the 13-mode test composite nitrogen oxides results.

Where:
DOB - Diesel Only-Baseline
DOC - Diesel Only-Converted
DFdown - Dual Fuel-Tubes down
DFup - Dual Fuel-Tubes up

The 13 mode composite NO_x emission results are compared in Figure 5.20. The composite NO_x results were almost identical for both diesel only configurations. The composite NO_x emissions were 17.5 g/kW.h and 17.6 g/kW.h before and after conversion, respectively. The lowest composite results of 11.8 g/kW.h were obtained during dual fuel operation with gas injection tubes pointed down. Dual fuel operation with tubes pointed up resulted in higher composite emissions, at 14.1 g/kW.h. However, they were still lower than the corresponding diesel only results.

5.2.6 Reactive Non-Methane Hydrocarbons (NMHC)

In diesel only modes, the brake specific NMHC emissions (measured as a hexane) were practically independent of brake power. Generally, they were at the same low level of about 0.5 g/kW.h across the entire brake power range. Only at low load modes, the NMHC emissions were slightly higher but never exceeded a level of 2 g/kW.h. The converted engine always produced lower NMHC emissions than the original stock engine for all testing points. Again, this agrees with the higher A/F ratios measured after conversion. In dual fuel modes, significantly higher NMHC emissions were measured at low and moderate load conditions compared to results from diesel only operation. At high load modes (above 75 kW), however, the NMHC emissions decreased rapidly and reached similar levels to diesel only results. The corresponding plots of the brake specific NMHC emissions as a function of brake power are shown in Figure 5.21. According to this figure, dual fuel operation with tubes pointed up produced the highest NMHC emissions over the entire brake power range.

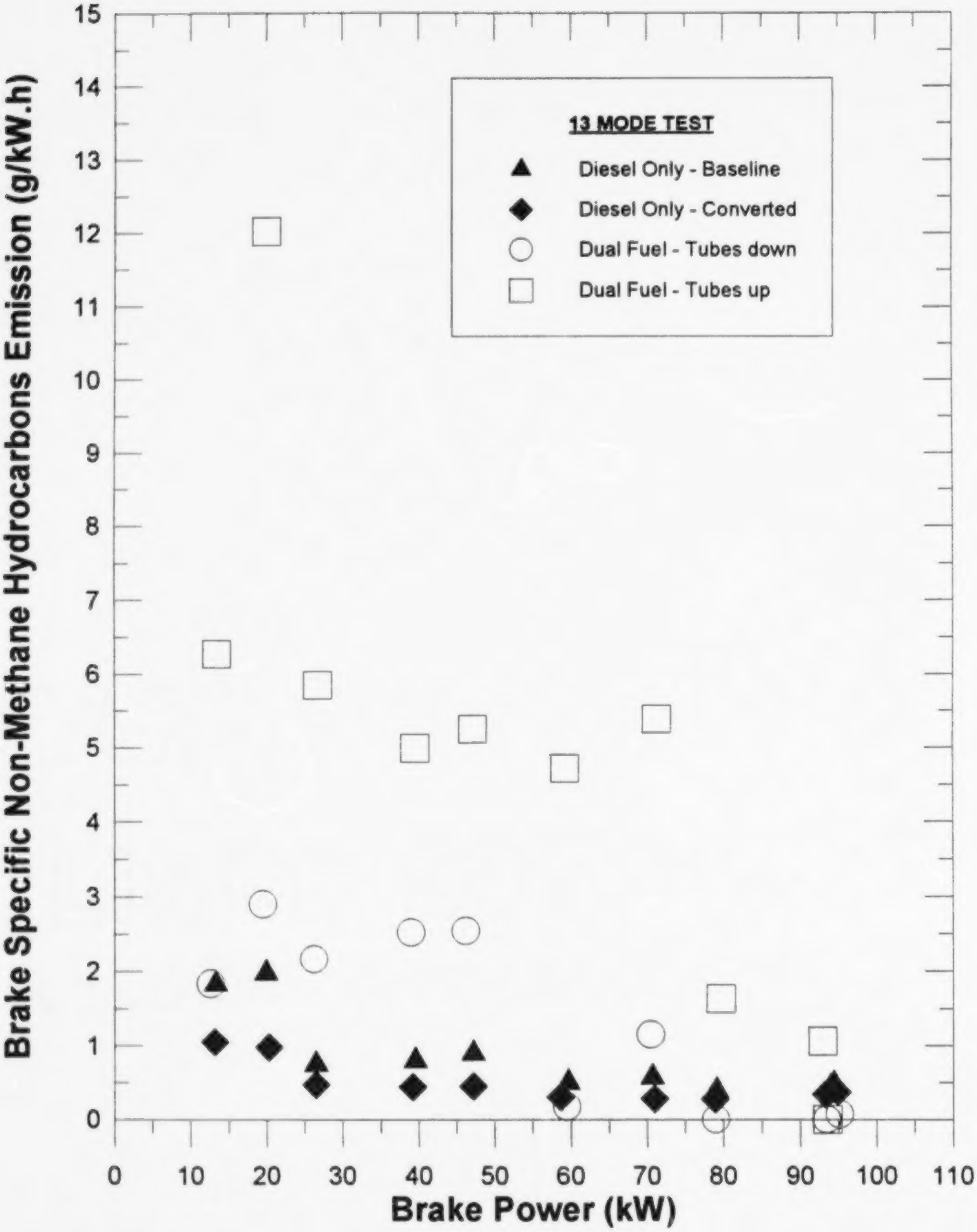


Figure 5.21 Brake specific non-methane hydrocarbons emissions in diesel only and dual fuel modes at part load (13-mode test).

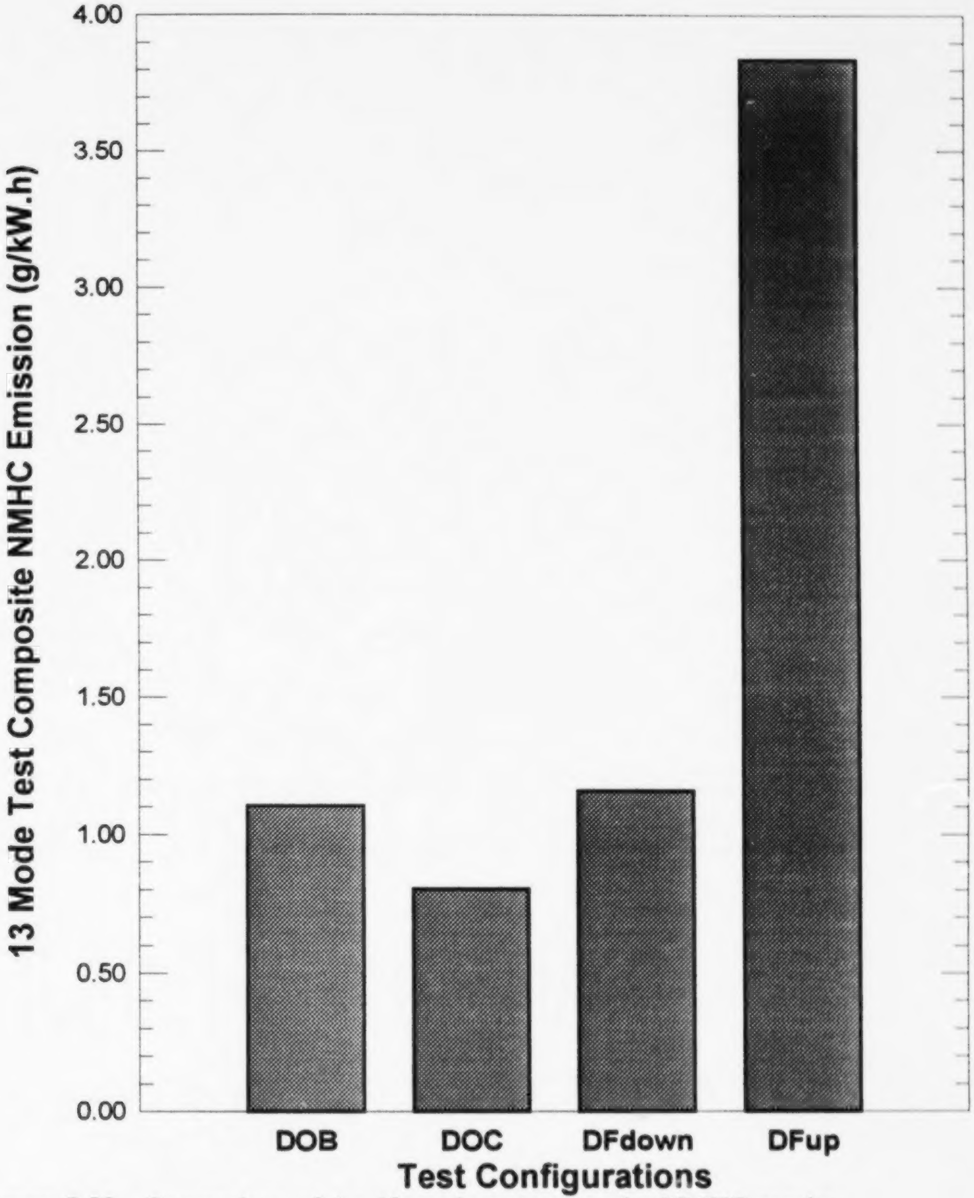


Figure 5.22 Comparison of the 13-mode test composite NMHC results.

Where:
DOB - Diesel Only-Baseline
DOC - Diesel Only-Converted
DFdown - Dual Fuel-Tubes down
DFup - Dual Fuel-Tubes up

The composite NMHC results are illustrated in Figure 5.22. On diesel only, the composite NMHC emission value was 1.10 g/kW.h before and 0.80 g/kW.h after conversion. This corresponds to 27 % reduction in NMHC emissions of the converted engine. On dual fuel, the composite NMHC results were slightly higher at 1.16 g/kW.h for tubes pointed down configuration, and at significantly higher level of 3.84 g/kW.h for tubes pointed up configuration. Since the NMHC was calculated as a difference of total hydrocarbon and methane hydrocarbon values, measurement accuracy was compromised in the dual fuel case.

5.2.7 Methane - Non Reactive Hydrocarbons (CH_4)

No methane fraction was found in the engine's exhaust gas during diesel only operation. On dual fuel, however, the brake specific methane emissions were high. As expected, the CH_4 emissions followed a similar pattern as diesel replacement ratio (Figure 5.13 and Figure 5.14) for both gas injection tubes configurations. They ranged from about 20 g/kW.h to about 60 g/kW.h. The maximum brake specific methane emissions were measured in mode 9 (about 60 kW at 1320 engine rpm). At very low and very high load conditions, smaller natural gas fuel fractions were used, resulting in a lower level of methane emissions (20-30 g/kW.h). As confirmed earlier in Section 5.1.8, natural scavenging losses were the primary source of methane emissions. No methane emissions were detected during modes 1, 4, and 13, which were run only on diesel fuel. The plots of the brake specific methane emissions as a function of brake power are shown in Figure 5.23.

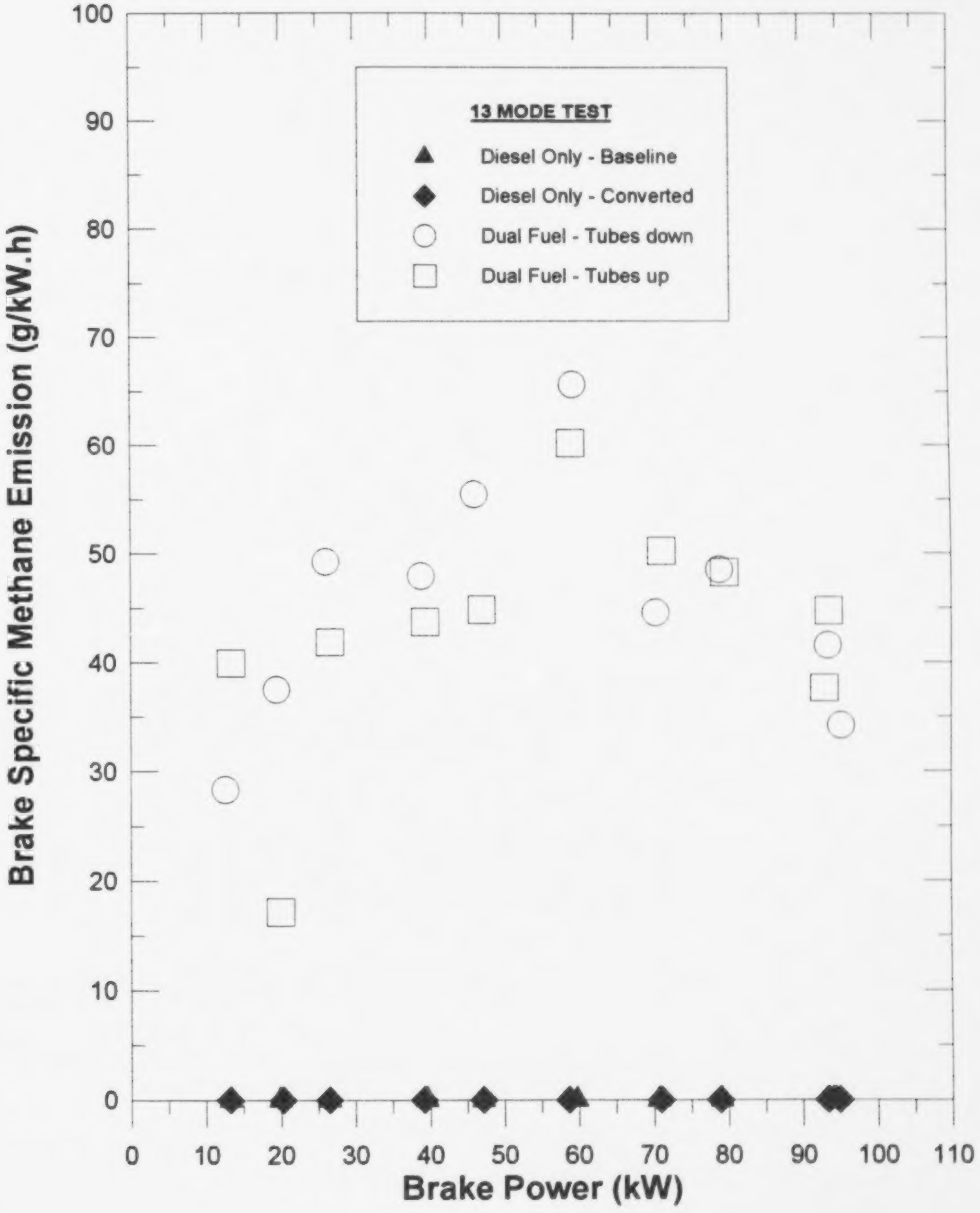


Figure 5.23 Brake specific methane emissions in diesel only and dual fuel modes at part load (13-mode test).

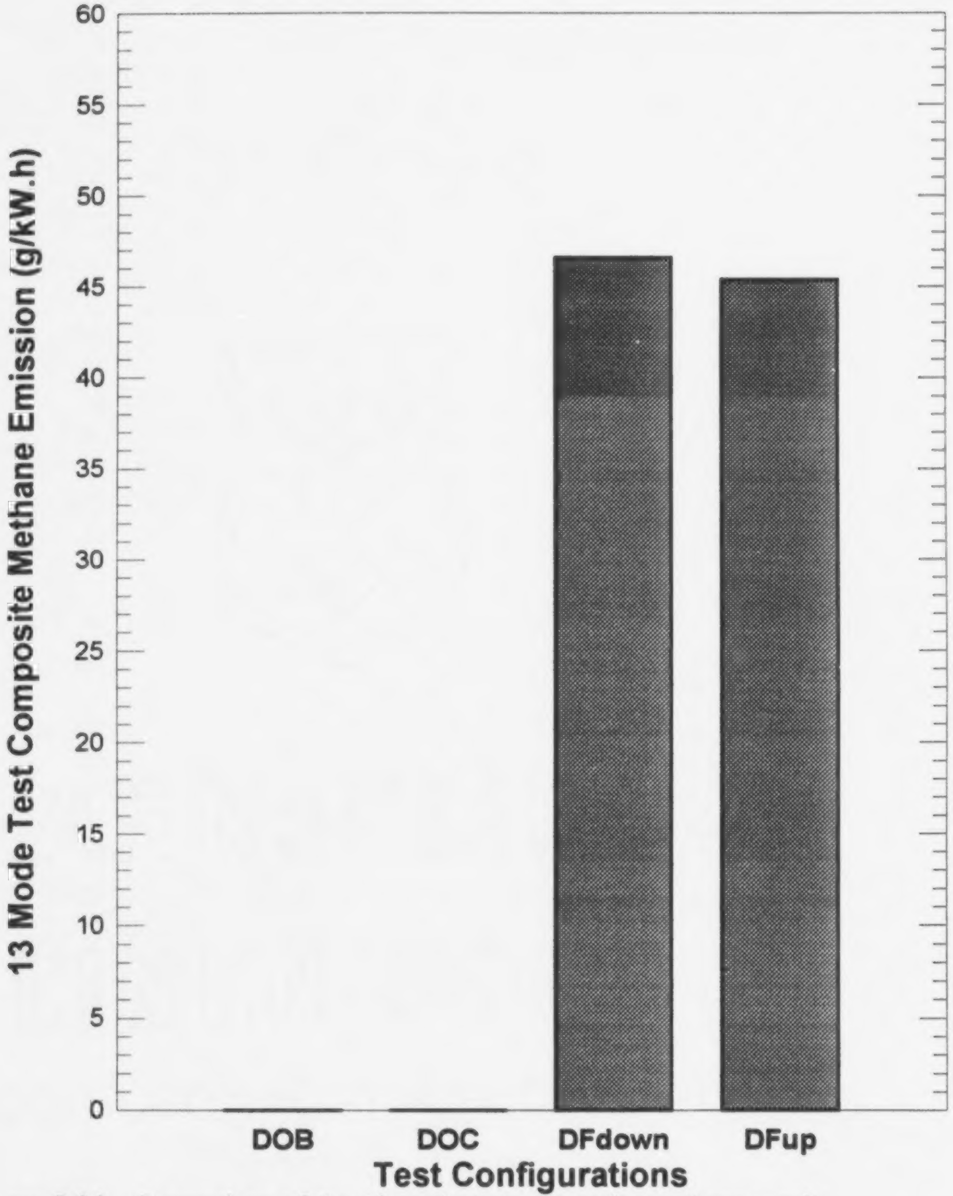


Figure 5.24 Comparison of the 13-mode test composite methane results.

Where:
DOB - Diesel Only-Baseline
DOC - Diesel Only-Converted
DFdown - Dual Fuel-Tubes down
DFup - Dual Fuel-Tubes up

The composite methane results obtained in both dual fuel modes are presented in Figure 5.24. Generally, the composite results were almost identical for both sets of tubes. They were at the level of 46.6 g/kW.h and 45.4 g/kW.h for the tubes pointed down and tubes pointed up configuration, respectively. This corresponds to about 33-34 % of the composite natural gas consumption (136-137 g/kW.h) in dual fuel modes.

5.2.8 Soot Emissions

The brake specific soot emissions as a function of brake power in diesel only and dual fuel modes are shown in Figure 5.25. As mentioned earlier, the main source of soot in the exhaust gas of diesel engine is incomplete combustion due to overfuelling or malfunction of the injection system. Generally, the tested engine was in a very good mechanical condition and emitted low levels of soot when operated on straight diesel. The converted engine, however, produced lower soot emissions over the entire power range than the original stock engine. This reduction in soot emissions agrees totally with slightly higher overall A/F ratios observed on the converted engine during straight diesel operation. A similar relationship was observed earlier for another product of incomplete combustion, carbon monoxide.

It is evident, from Figure 5.25, that dual fuel operation with tubes pointed down produced similar soot emission levels to the corresponding diesel only levels. In contrast, the soot emission levels were significantly higher for all testing points during dual fuel operation with tubes pointed up. The increased soot formation in the latter case was probably due to longer ignition delay of diesel pilot [24, 33]. Consequently, there was

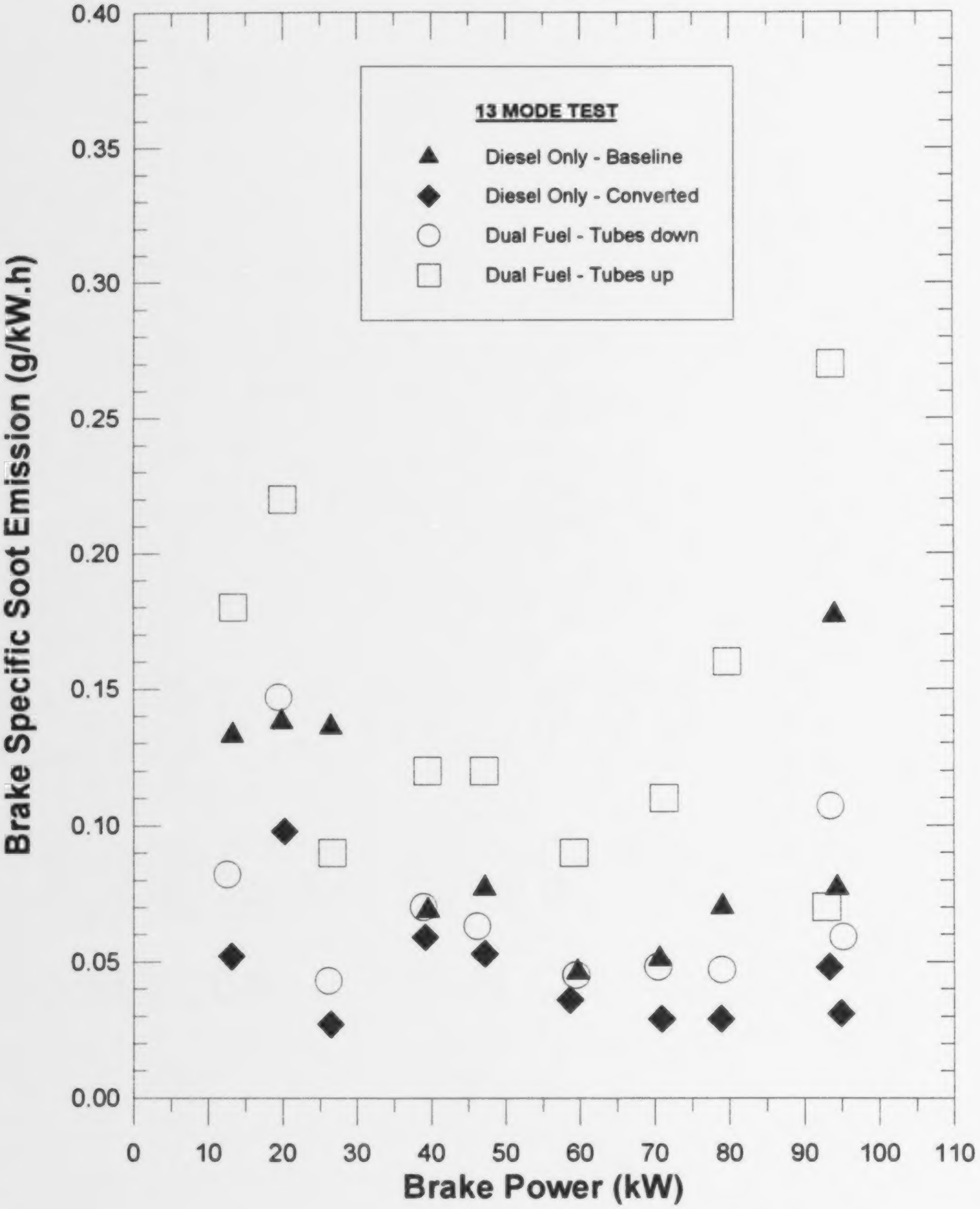


Figure 5.25 Brake specific soot emissions in diesel only and dual fuel modes at part load (13-mode test).

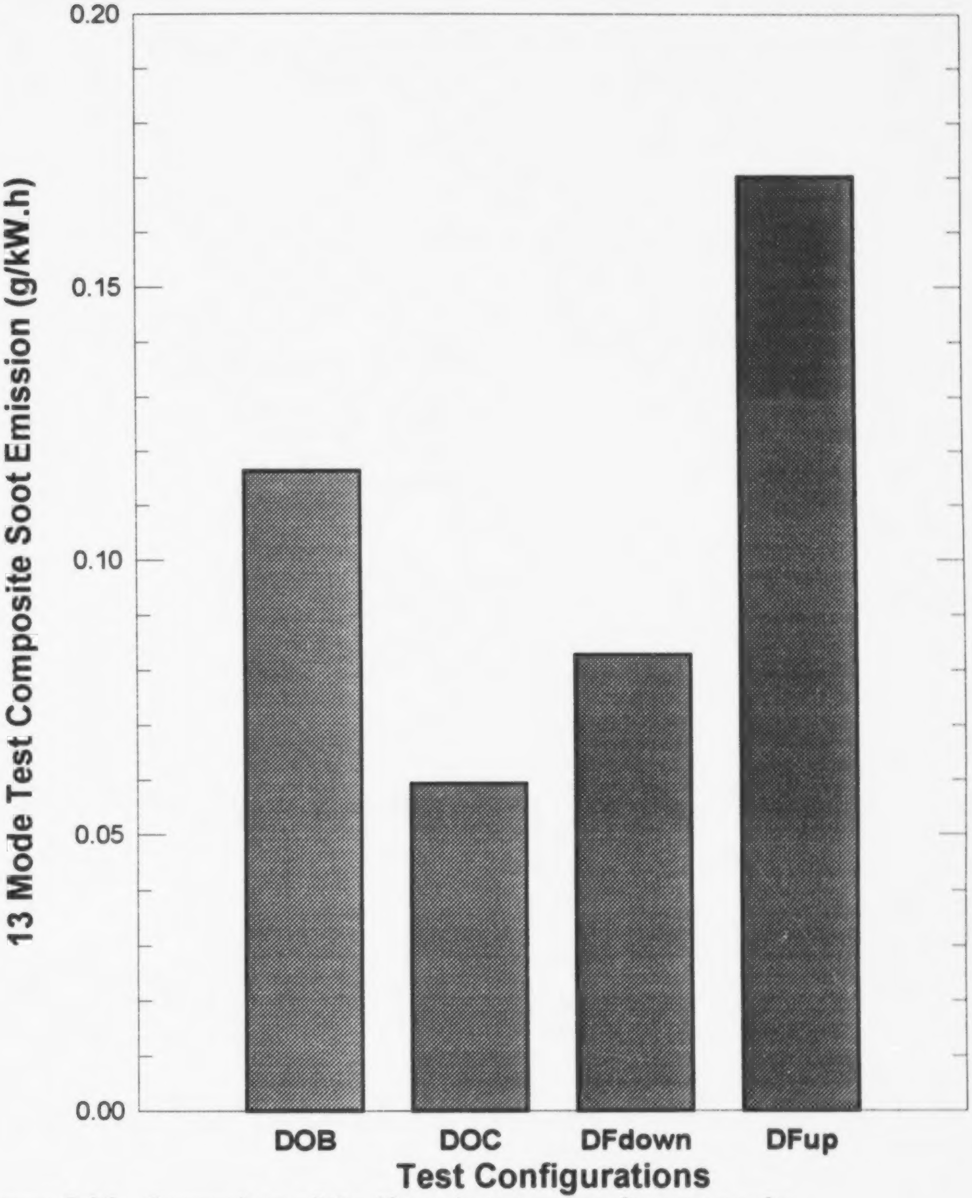


Figure 5.26 Comparison of the 13-mode test composite soot results.

Where:
DOB - Diesel Only-Baseline
DOC - Diesel Only-Converted
DFdown - Dual Fuel-Tubes down
DFup - Dual Fuel-Tubes up

no sufficient time for diesel fuel to be burned out completely during the remainder of the combustion process. Like in the case of the CO emissions, the soot plots for both dual fuel modes follow a similar pattern: the soot emissions were relatively high at low power modes (less than 25 kW), relatively low at moderate power modes (25-70 kW), and relatively high again at high power modes (above 70 kW).

The composite soot emissions for all test configurations are shown in Figure 5.26. For diesel only operation, the composite soot emissions were 0.116 g/kW.h before conversion and 0.059 g/kW.h after conversion. This corresponds to a 49 % reduction in soot emissions of the converted engine over the stock one. During dual fuel operation with tubes pointed down, the composite soot results were at the level of 0.083 g/kW.h, just between the corresponding diesel only results obtained before and after conversion. The highest composite soot results were at the level of 0.170 g/kW.h when engine ran in dual fuel mode with tubes pointed up. These results again show that tubes pointed down performed better, producing less than half of the soot emissions obtained with tubes pointed up.

5.3 COMPARISON OF THE RESULTS OF THE FOUR DIFFERENT TEST CONFIGURATIONS.

As noted earlier, both the maximum torque and the Japanese 13-mode test were conducted for the following four test configurations:

1. Diesel only operation of the original stock engine (baseline test)
2. Diesel only operation of the converted engine
3. Dual fuel operation with natural gas injection tubes pointed down
4. Dual fuel operation with natural gas injection tubes pointed up

Tables 5.1 and 5.2 compare the performance and emission results obtained for the various test configurations during both the maximum torque and multi-mode tests. The results of the original stock engine (diesel only baseline) are used as a standard for rating the results of the other three test configurations. In the case of diesel replacement ratio and methane emissions, the results of dual fuel operation with tubes pointed down are used as a standard. Table 5.1 compares the peak power, torque and thermal efficiency in the maximum torque test, while table 5.2 compares the composite fuel consumption and emission results obtained during the Japanese 13-mode test.

5.3.1 Diesel Only Operation - Baseline

The baseline diesel only configuration produced the highest torque. In general, however, the maximum torque and power results were very similar for all test configurations. High thermal efficiencies, typical for a diesel engine, were observed while the composite carbon dioxide emissions were found to be the highest. The baseline engine did not produce much smoke; thus, the composite soot results were low. The composite non-methane hydrocarbon emissions results were also found to be low.

5.3.2 Diesel Only Operation - After Conversion

The highest thermal efficiencies were observed during diesel only operation of the converted engine at the maximum torque conditions. The power and torque output matched the power and torque characteristics of the stock engine very well. The converted engine also performed very well on diesel fuel only during the multi-mode test. This was confirmed by the composite fuel consumption results which were found to be the lowest. Additionally, this configuration produced the lowest composite carbon monoxide, non-methane hydrocarbons and soot results. The composite NO_x results were slightly higher (less than 1 %) than the corresponding baseline diesel results.

5.3.3 Dual Fuel Operation with Gas Injection Tubes Pointed Down

The highest power output (about 4 % higher than the baseline configuration) was measured during the maximum torque test for this test configuration. Generally, the converted engine operated in dual fuel mode with tubes pointed down matched well the power and torque output of the stock engine. The thermal efficiencies were significantly lower due to the natural gas scavenging losses. The lowest composite CO_2 (7.7 % reduction) and NO_x (32.5 % reduction) results were measured for this configuration. The composite soot results were about 30 % lower while the composite CO emissions were significantly higher (279 %) and NMHC results were slightly higher (4.8 %) than the corresponding baseline results. Because of the natural gas scavenging losses, the composite total fuel consumption increased about 19 % and high methane concentrations were observed in the exhaust gas. The dual fuel operation with tubes pointed down replaced about 24 % of diesel fuel over the 13 mode procedure.

5.3.4 Dual Fuel Operation with Gas Injection Tubes Pointed Up

As in the case of other test configurations, the converted engine operated in dual fuel mode with tubes pointed up matched well the power and torque output of the stock engine. The thermal efficiencies were significantly lower due to the natural gas scavenging losses. The composite CO_2 results were 5.6 % lower while the composite NO_x results decreased 19.2 %. The highest composite CO, NMHC, and soot results were measured for this configuration. The composite total fuel consumption increased 20.5 % and was at the highest level. The composite methane emissions were high, but slightly lower (2.6 %) than methane results obtained during dual fuel operation with the tubes pointed down configuration. The diesel replacement ratio was 5.8 % lower than the corresponding level for tubes pointed down.

Table 5.1 Comparison of the peak power, torque, and thermal efficiency results obtained during the maximum torque test for the four test configurations.

	Diesel Only Baseline (%)	Diesel Only Converted (%)	Dual Fuel Tubes down (%)	Dual Fuel Tubes up (%)
Peak Power	100	96.8	103.8	98.9
Peak Torque	100	98.9	97.2	99.6
Thermal Efficiency	100	102.5	86.8	89.0

Table 5.2 Comparison of the composite fuel consumption and emission results obtained during the Japanese 13-mode test for the four test configurations.

	Diesel Only Baseline (%)	Diesel Only Converted (%)	Dual Fuel Tubes down (%)	Dual Fuel Tubes up (%)
Total Fuel Consumption	100	96.8	119.0	120.5
Diesel Replacement	N/A	N/A	24.1	22.7
CO_2	100	98.4	92.3	95.4
CO	100	74.0	278.9	481.4
NO_x	100	100.8	67.5	80.8
NMHC	100	72.6	104.8	347.4
CH_4	N/A	N/A	100	97.4
SOOT	100	50.9	71.1	146.1

CHAPTER 6 SUMMARY AND CONCLUSIONS

6.1 SUMMARY

The objective of the present study was to develop a prototype conversion system to convert a Detroit Diesel 6V-71 transit bus engine to dual fuel natural gas/diesel operation, and to evaluate the performance characteristics in both diesel only and dual fuel modes. The AFS Electronic Dual Fuel Injection System was used for converting this diesel engine to natural gas/diesel operation. To make the conversion system simple and affordable, minimal changes were done to the engine. The existing compression ratio, blower drive and diesel injection system were retained as much as possible. The existing diesel fuel injectors were used to inject pilot diesel as controlled by an electronic rack actuator. The gaseous fuel was delivered to each cylinder by individual high speed solenoid valves which controlled the timing and volume of injected natural gas. The gas valves were installed in two special manifolds mounted on the outside of the engine block and delivered gas through tubes to each cylinder intake port. Two different sets of gas injection tubes were used. The first set of tubes was deflected 30° downwards and directed the natural gas down towards the piston while the second set of tubes was deflected 30° upwards and directed the natural gas up towards the top of the cylinder.

The conversion kit was optimized during the process of engine mapping. Engine mapping involved operating the engine at a fixed speed and load conditions and changing diesel fuel and natural gas fuel quantities as well as natural gas injection timing to determine their optimum values. It was found during the process of optimization that diesel pilot quantity had a significant effect on total hydrocarbon emissions during dual fuel operation. When the diesel pilot was below a threshold level of 11 mm^3/injection, the combustion of the natural gas failed to occur and high methane emissions were observed. Consequently at idle and very low loads the engine was set to operate on straight diesel. Another complication, the knock phenomenon, was encountered at high load conditions as relatively rich mixtures of natural gas with air were employed. To avoid knock, the diesel pilot was increased simultaneously with reducing natural gas injection rate.

To compare the diesel only performance and emissions of the stock engine with those of the converted engine, engine was tested using the maximum torque procedure and the Japanese 13-mode procedure before and after the conversion system was installed. The 13-mode test schedule is a standard test which involved running the engine at 13 speed/load points representing typical operating conditions. During the maximum torque test the engine ran at maximum "throttle" over a range of engine speeds from 800 rpm to 2200 rpm in 200 rpm increments. The standard engine was tested in diesel only mode while the converted engine was tested in both diesel only and dual fuel modes for each test regime. On dual fuel, the engine performance and emissions were investigated for the two different sets of gas injection tubes. All tests were carried out on an eddy-current type dynamometer.

The converted engine maintained its original power and torque levels. The shape of the power/speed and torque/speed curves of the converted engine operated either on diesel only or on dual fuel were practically the same as those of the stock engine. The peak power values varied between 127 kW (for diesel only operation after conversion) and 136 kW (for dual fuel operation with tubes pointed down). There was even smaller variation in the peak torque results which ranged between 736 N.m (for dual fuel operation with tubes pointed down) and 758 N.m (for baseline diesel operation). For all test configurations, the peak torque was obtained at the same engine speed of 1000 rpm.

For maximum torque operation, the b.s.f.c. was higher in dual fuel modes than in straight diesel modes across the entire engine speed range. This was mainly due to natural gas scavenging losses. The b.s.f.c. averaged 279 g/kW.h (baseline engine) and 267 g/kW.h (after conversion) on straight diesel, while the average total b.s.f.c. on dual fuel was 313 g/kW.h (tubes down) and 324 g/kW.h (tubes up). Similarly, the composite fuel consumption over the 13-mode test was higher for dual fuel operation. In diesel only mode, the composite results were 313 g/kW.h and 303 g/kW.h before and after conversion, respectively. On dual fuel, the corresponding diesel equivalent composite results were 373 g/kW.h (tubes down) and 378 g/kW.h (tubes up). Because of the natural gas scavenging losses, the total energy consumption increased about 23 % and 26 % for tubes deflected down and tubes deflected up, respectively.

High air charge temperature from the blower and high compression ratio (18.7:1) limited the amount of diesel fuel that could be displaced by natural gas without the onset of knock. The highest diesel fuel replacement ratio of 45.2 % (tubes down) and of 46.6 % (tubes up) was observed at 800 rpm during the maximum torque test. During the 13-mode test, the most heavily weighted modes (1,4, and 13) were operated only on diesel fuel. Consequently, composite diesel replacement ratios as low as 24.1 % and 22.7 % were obtained for tubes pointed down and up, respectively.

As expected for natural gas fuelling, the brake specific CO_2 emissions were reduced in dual fuel modes. The average brake specific CO_2 emissions, measured over the maximum torque procedure, were 810 g/kW.h on straight diesel, and 730 g/kW.h on dual fuel. This corresponded to about 10 % reduction in CO_2 emissions for dual fuel operation. Similarly, the composite CO_2 emission results over the 13-mode test were lower for dual fuel operation. The highest composite CO_2 emissions (963 g/kW.h) were produced by the standard engine on straight diesel. The composite CO_2 results of the converted engine on straight diesel were slightly lower, at the level of 948 g/kW.h. On dual fuel, the composite CO_2 results were further reduced to the level of 919 g/kW.h for tubes pointed up and to the level of 890 g/kW.h for tubes pointed down. The reduction in the CO_2 emissions was directly related to the lower carbon content of natural gas fuel.

In general, the engine produced low levels of CO emissions when operated on straight diesel. The composite CO emissions were 6.1 g/kW.h before conversion and 4.5 g/kW.h after conversion. This corresponded to 26 % reduction in CO emissions of the converted engine over the standard one. On dual fuel, the CO emissions were about 3-5 times higher compared to diesel only operation of the standard stock engine. The composite results were 16.9 g/kW.h and 29.2 g/kW.h for tubes pointed down and for tubes pointed up configurations, respectively. These results together with the maximum torque results proved that tubes pointed down performed better, producing only about 50 % of the CO emissions obtained with tubes pointed up. Generally, the CO emission levels were acceptable low. Only the composite CO result obtained during dual fuel operation with tubes pointed up exceeded 1994 EPA standard of 20.8 g/kW.h.

On diesel only, the composite NMHC emission value was 1.10 g/kW.h before and 0.80 g/kW.h after conversion. This corresponded to 27 % reduction in NMHC emissions of the converted engine. On dual fuel, the composite NMHC results were slightly higher at 1.16 g/kW.h for the tubes pointed down configuration, and significantly higher at 3.84 g/kW.h for the tubes pointed up configuration. As in the case of the CO emissions, the hydrocarbon emissions are low. Only the composite NMHC result obtained during dual fuel operation with tubes pointed up exceeded the 1994 EPA standard of 1.7 g/kW.h.

There were no methane emissions during diesel only operation, while on dual fuel high concentrations of methane were measured in the exhaust gas. However, methane is not considered to be a serious pollutant since it is non-reactive in the formation of ozone. Natural gas scavenging losses were the primary source of methane emissions. The composite results were almost identical for both sets of tubes. They were at 46.6 g/kW.h and 45.4 g/kW.h for tubes pointed down and up configuration, respectively. This corresponded to about 40 % of the composite methane consumption (116.4 and 116.9 g/kW.h) in dual fuel modes.

For diesel only operation, the composite soot emissions were 0.116 g/kW.h before conversion and 0.059 g/kW.h after conversion. This corresponds to 49 % reduction in soot emissions of the converted engine over the stock one. During dual fuel operation with tubes pointed down, the composite soot results were at the level of 0.083 g/kW.h. The highest composite soot results were at the level of 0.170 g/kW.h when engine ran in dual fuel mode with tubes pointed up. These results confirmed that tubes pointed down performed better, producing less than half of the soot emissions obtained with tubes pointed up.

The composite NO_x results were almost identical for diesel only operation both before and after conversion. The composite NO_x emissions were 17.5 g/kW.h and 17.6 g/kW.h before and after conversion, respectively. Dual fuel operation resulted in lower NO_x emissions. The lowest composite result of 11.8 g/kW.h was obtained during dual fuel operation with gas injection tubes pointed down. This composite result, however, was far above the 1994 Japanese NO_x emission limit of 6.0 g/kW.h and the 1994 EPA standard of 6.7 g/kW.h.

6.2 CONCLUSIONS

The results from the study show that a standard Detroit Diesel 6V-71 transit bus two stroke engine can be converted to dual fuel natural gas/diesel operation. The Electronic Dual Fuel Injection System of Alternative Fuel Systems Inc. was used as the basis for the conversion. The electronically controlled, multi-point natural gas injection system and diesel rack positioning system performed satisfactorily over the entire operating range of the 6V-71 engine. The conversion added flexible, mapped control of both natural gas and diesel injection. The converted engine was designed to operate interchangeably on natural with diesel pilot, or on 100 % diesel fuel. Thus, full diesel operation can be kept in event of gas supply failure. None of the AFS system components caused any technical problems during the period of testing.

The converted engine maintained its original power and torque levels in both dual fuel and diesel only operation. On diesel fuel, the converted engine consumed less fuel and produced lower CO_2, CO, NMHC, and soot composite results than a standard engine. During dual fuel operation with tubes pointed down, the composite emissions of CO_2, NO_x, and soot were lower, while the composite emissions of CO, total hydrocarbons, and non-methane hydrocarbons were higher when compared with the standard engine. The specific diesel fuel consumption was reduced about 24 % over the 13 mode procedure. When natural gas was injected upward, the composite emissions of CO, and soot were significantly higher and less diesel fuel was displaced by natural gas (about 22.7 %). It was found in this study that engine knock limited the amount of natural gas that could be used to displace diesel fuel. The highest diesel fuel replacement ratios (45-47 %) were observed at low engine speeds and maximum torque conditions. Due to gas injection during the scavenging process, high natural gas losses were encountered on dual fuel.

6.3 SUGGESTED IMPROVEMENTS

The inefficiency during dual fuel operation offset the potential fuel cost savings attainable from the lower price of natural gas. Therefore, improvements are needed to reduce the natural gas scavenging losses and to increase diesel fuel replacement ratio before any real consideration could be given to making actual conversion in the field. Scavenging losses can be reduced by injecting the gaseous fuel through the cylinder liner wall above the intake ports, close to the cylinder head. This operation would require, however, that an engine be disassembled and a hole be drilled in the cylinder liners and in the engine block. Injection through the upper part of the cylinder wall would offer more flexibility in terms of time available for injection (the exact amount of time available depends on the location of the nozzle relative to the cylinder head). Since the natural gas could be injected after the intake ports were covered by the piston and the exhaust valves fully closed, the scavenging losses would be avoided. Another way to reduce the amount of scavenged natural gas is to shorten the period of time between the end of natural gas injection and exhaust valve closure. This can be done by camshaft modification but it would affect engine scavenging performance.

The knock margin of the engine can be significantly widened by lowering the engine's compression ratio and by adding an air charge aftercooler to the intake system. Lower compression ratio and higher natural gas utilization would lower NO_x emission levels substantially and allow to make system economic. In case the CO and NMHC emission levels become too high, installation of an oxidation catalyst would be recommended.

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APPENDIX A

ENGINE MAPPING RESULTS

Table A1 Diesel fuel pump calibration map.

Diesel fuel volume as a function of engine speed and fuel rack position. (mm^3/injection)

Rack mm	500 rpm	550 rpm	560 rpm	600 rpm	800 rpm	1000 rpm	1200 rpm	1400 rpm	1800 rpm	2200 rpm	2300 rpm
9.15	9.68	X	X	11.90	12.24	13.54	12.87	12.86	X	X	X
10.15	16.30	X	X	20.03	19.52	19.06	17.87	18.62	17.87	X	X
11.15	X	23.99	X	24.68	23.73	25.80	25.93	24.89	22.55	22.00	22.06
12.15	X	31.19	X	33.37	32.33	33.14	32.27	31.60	28.57	27.30	26.92
13.15	X	44.48	X	42.70	40.17	40.38	38.83	37.39	34.93	33.77	32.94
14.15	X	52.89	X	50.45	48.18	48.21	46.37	44.72	41.24	40.63	39.10
15.15	X	61.39	X	59.47	56.88	55.26	52.95	52.15	47.18	44.78	44.56
16.15	X	X	69.90	68.21	64.33	62.85	60.66	58.35	52.99	50.11	49.88
17.15	X	X	66.24	74.98	69.84	67.35	65.60	61.96	57.02	52.53	50.34
18.15	X	X	78.27	77.56	74.14	70.56	68.20	66.18	59.44	54.40	53.75

Table A2 Pilot diesel map for dual-fuel operation (mm^3/injection).

Total Commanded Fuel (mm^3/inj)	ENGINE SPEED COLUMNS (rev/min)									
	0	500	550	700	800	1000	1400	1800	2200	2300
0	0	0	0	0	0	0	0	0	0	0
5	5	5	5	5	5	5	5	5	5	5
10	10	10	10	10	10	10	10	10	10	10
20	20	20	12	12	12	12	15	20	20	20
30	30	30	15	15	15	12	20	20	20	30
40	40	30	20	20	20	20	20	20	30	40
50	50	30	20	20	20	20	20	30	40	50
60	60	30	20	20	20	20	25	35	50	60
70	70	30	20	20	20	33	35	45	60	70
90	80	30	30	30	35	43	45	50	70	80

Table A3 Diesel fuel injection pump lever position map (mm).

Total Commanded Fuel (mm^3/inj)	ENGINE SPEED COLUMNS (rev/min)									
	0	545	550	600	800	1000	1400	1800	2200	2300
0	7.40	7.40	7.40	7.50	7.70	7.50	6.90	7.00	7.10	7.10
1	7.55	7.51	7.51	7.51	7.71	7.51	6.91	7.01	7.11	7.11
10	9.00	8.80	8.80	8.80	9.00	8.98	8.70	8.80	9.00	9.00
20	10.40	10.20	10.20	10.20	10.20	10.20	10.40	10.70	10.75	10.75
30	12.20	12.00	12.00	11.80	11.90	11.70	11.90	12.40	12.55	12.60
40	13.00	12.80	12.80	12.80	13.20	13.20	13.50	13.90	14.10	14.40
50	14.00	13.80	13.80	14.10	14.30	14.30	14.80	15.70	16.10	16.10
60	15.50	15.00	15.00	15.20	15.60	15.70	16.60	17.30	17.20	17.20
70	16.50	16.15	16.15	16.40	17.20	17.70	17.70	17.70	17.70	17.70
80	17.70	17.70	17.70	17.70	17.70	17.70	17.70	17.70	17.70	17.70

Table A4 Natural gas valve timing map for tubes pointed upwards. End of injection BTDC.

Natural Gas Flow (mg/inj)	ENGINE SPEED COLUMNS (rev/min)									
	0	500	550	700	800	1000	1400	1800	2200	2300
0	150	150	150	150	150	150	150	150	150	150
5	150	150	150	150	150	150	150	150	150	150
10	150	150	150	150	150	150	150	150	145	145
20	145	140	140	140	140	140	140	135	135	132.5
30	140	140	140	140	137.5	130	125	120	120	117.5
40	130	130	130	130	127.5	130	105	105	105	105
50	130	130	130	125	120	115	105	105	105	105
60	130	130	125	120	115	110	102.5	102.5	102.5	102.5
70	130	130	125	120	115	110	102.5	102.5	102.5	102.5
80	130	130	125	120	115	110	102.5	102.5	102.5	102.5

Table A5 Natural gas valve timing map for tubes pointed downwards. End of injection BTDC.

Natural Gas Flow (mg/inj)	ENGINE SPEED COLUMNS (rev/min)									
	0	500	550	700	800	1000	1400	1800	2200	2300
0	170	170	170	170	170	170	170	170	170	170
5	170	170	170	170	170	170	170	170	165	165
10	170	170	170	165	165	170	160	155	145	145
20	160	160	160	160	160	155	155	140	140	140
30	155	155	155	152.5	152.5	160	140	140	140	140
40	145	145	145	145	145	150	130	130	130	130
50	145	145	145	145	145	150	130	130	130	130
60	145	145	145	145	145	150	130	130	130	130
70	145	145	145	145	145	150	130	130	130	130
80	145	145	145	145	145	150	130	130	130	130

Table A6 Maximum Fuel Function for diesel only and dual fuel operation.

Engine Speed (rev/min)	Maximum Fuel Function Diesel Fuel Mode (mm^3/inj)	Maximum Fuel Function Dual Fuel Mode (diesel equivalent mm^3/inj)
0	60.0	60.0
500	72.0	72.0
600	75.0	75.0
800	77.2	86.0
1000	72.9	82.0
1200	69.0	79.0
1400	67.3	78.0
1600	62.5	70.0
1800	66.0	66.0
2000	58.2	64.0
2200	55.5	59.0
2300	50.0	50.0
2400	0	0

Table A7 Natural gas injection valve calibration data.
Calibration pressure 2070 kPa (300 psi).

ENERGIZE TIME (ms)	NATURAL GAS FLOW (mg/injection)
3.58	14.50
6.66	33.59
7.63	38.20
13.00	69.80

APPENDIX B

THE MAXIMUM TORQUE TEST RESULTS

Table B1 Fuel consumption and raw emissions at maximum torque. Diesel only operation - baseline (DOB).

Speed rpm	Tq Nm	NG g/s	Diesel g/s	Air g/s	O2 %	CO %	CO2 %	NOx ppm	THC ppm C6	CH4 ppm C	RHC ppm C6	Smk HSU
796	705	0	5.11	123	9.1	1.017	6.81	688	17	0	17	32
997	733	0	6.04	158	9.3	1.016	7.04	760	13	0	13	17
1196	717	0	6.92	190	9.4	0.838	7.03	783	13	0	13	22
1397	710	0	7.80	222	9.7	0.414	6.83	878	12	0	12	11
1597	681	0	8.27	251	9.9	0.244	6.74	923	14	0	14	8
1797	638	0	9.04	282	10.1	0.193	6.68	931	16	0	16	10
1998	603	0	9.63	313	10.5	0.096	6.49	981	17	0	17	6
2198	561	0	10.14	345	10.7	0.059	6.31	998	19	0	19	5

Table B2 Corrected torque, corrected power, tailpipe temperature, A/F ratio, carbon balance, H/C ratio, and thermal efficiency at maximum torque. Diesel only operation - baseline (DOB).

Speed rpm	Tqcor Nm	Powcor kW	TailTem deg C	A/F mass	A/F emm	Cbal in/out %	H/C emm	Eff %
796	733.3	61.1	283	24.00	26.00	114	2.8	26.6
997	758.1	79.1	381	26.20	25.80	101	2.4	29.3
1196	739.7	92.7	422	27.40	26.20	99	2.5	30.1
1397	729.9	106.8	440	28.40	28.10	104	2.8	30.8
1597	696.8	116.5	450	30.40	29.00	101	2.9	31.9
1797	651.4	122.6	434	31.20	29.40	100	2.9	30.8
1998	613.7	128.4	440	32.50	30.60	100	2.9	30.4
2198	569.2	131.0	462	34.00	31.40	99	2.9	29.5

Table B3 Observed torque, observed power, and emissions in mg/s (g/s) at maximum torque.
Diesel only operation - baseline (DOB).

Speed rpm	Tq Nm	Pow kW	H2O g/s	O2 mg/s	CO mg/s	CO2 mg/s	NOx mg/s	THC mg/s	CH4 mg/s	RHC mg/s	Soot mg/s
796	705	58.7	8.0	11946	1167.0	12277	129.7	6.1	0	6.1	14.55
997	733	76.5	9.3	15697	1502.7	16363	184.5	6.1	0	6.1	9.90
1196	717	89.9	11.3	18941	1484.0	19563	227.8	7.0	0	7.0	15.36
1397	710	103.8	13.5	22947	855.5	22163	297.9	7.8	0	7.8	8.96
1597	681	114.0	15.1	26548	570.5	24736	354.3	9.9	0	9.9	7.37
1797	638	120.1	16.7	30265	507.4	27556	401.3	12.7	0	12.7	10.34
1998	603	126.3	17.9	34961	279.2	29810	470.7	15.2	0	15.2	6.88
2198	561	129.2	19.9	39031	188.4	31976	525.6	18.5	0	18.5	6.30

Table B4 Brake specific fuel consumption and brake specific emissions in g/kW.h at maximum torque.
Diesel only operation - baseline (DOB).

Speed rpm	bsfcD g/kWh	bsfcNG g/kWh	bsO2 g/kWh	bsCO g/kWh	bsCO2 g/kWh	bsNOx g/kWh	bsTHC g/kWh	bsCH4 g/kWh	bsRHC g/kWh	bsSoot g/kWh
796	313.3	0	732.2	71.52	752.5	7.95	0.37	0	0.37	0.892
997	284.2	0	739.1	70.75	770.4	8.69	0.28	0	0.28	0.466
1196	277.1	0	758.7	59.44	783.6	9.12	0.28	0	0.28	0.615
1397	270.7	0	796.0	29.67	768.8	10.33	0.27	0	0.27	0.311
1597	261.4	0	838.7	18.02	781.5	11.19	0.31	0	0.31	0.233
1797	271.0	0	906.9	15.20	825.7	12.02	0.38	0	0.38	0.310
1998	274.7	0	996.9	7.96	850.0	13.42	0.43	0	0.43	0.196
2198	282.5	0	1087.4	5.25	885.9	14.64	0.51	0	0.51	0.176

Table B5 Fuel consumption and raw emissions at maximum torque.
Diesel only operation - after conversion (DOC).

Speed rpm	Tq Nm	NG g/s	Diesel g/s	Air g/s	O2 %	CO %	CO2 %	NOx ppm	THC ppm C6	CH4 ppm C	RHC ppm C6	Tr %
805	717	0	4.619	118	10.2	1.122	7.01	1065	14	0	14	99.01
1001	735	0	5.674	154	10.6	0.547	6.93	1142	11	0	11	99.41
1201	716	0	6.378	187	10.7	0.459	6.88	1061	16	0	16	99.30
1401	698	0	7.254	219	10.9	0.287	6.91	1250	9	0	9	99.44
1602	656	0	8.065	249	11.0	0.231	6.78	1152	17	0	17	99.45
1803	624	0	8.829	279	11.1	0.173	6.72	1209	9	0	9	99.17
2003	598	0	9.442	312	11.3	0.094	6.65	1298	4	0	4	99.47
2203	540	0	9.868	342	11.7	0.067	6.33	1258	7	0	7	99.50

Table B6 Corrected torque, corrected power, air box temperature, tailpipe temperature, A/F ratio, carbon balance, H/C ratio, and thermal efficiency at maximum torque.
Diesel only operation - after conversion (DOC).

Speed rpm	Tqcor Nm	Powcor kW	BoxTem deg C	TailTem deg C	A/F mass	A/F emm	Cbal in/out %	H/C emm	Eff %
805	731.6	61.7	66.5	412	25.70	26.34	101	1.8	30.4
1001	749.9	78.6	67.6	405	27.11	28.33	104	2.0	31.5
1201	729.3	91.7	72.6	426	29.29	28.84	98	2.0	32.7
1401	709.6	104.1	80.8	407	30.16	29.44	97	2.0	32.7
1602	666.1	111.7	86.7	410	30.87	30.07	98	2.0	31.6
1803	632.3	119.3	95.0	410	31.63	30.58	97	2.1	30.9
2003	605.1	126.9	101.4	419	33.00	31.22	95	2.1	30.8
2203	545.3	125.8	108.8	425	34.64	32.79	95	2.1	29.3

Table B7 Observed torque, observed power, and emissions in mg/s (g/s) at maximum torque.
Diesel only operation - after conversion (DOC).

Speed rpm	Tq Nm	Pow kW	H2O g/s	02 mg/s	CO mg/s	CO2 mg/s	NOx mg/s	THC mg/s	CH4 mg/s	RHC mg/s	Soot mg/s
805	717	60.4	5.2	13180	1264.8	12415	197.0	4.8	0	4.8	1.73
1001	735	77.0	7.0	17608	797.8	15872	273.4	4.9	0	4.9	1.34
1201	716	90.1	8.3	21666	811.5	19101	308.1	8.7	0	8.7	1.91
1401	698	102.4	9.5	25693	594.2	22469	425.0	5.8	0	5.8	1.79
1602	656	110.0	10.9	29606	543.9	25071	445.6	12.6	0	12.6	1.99
1803	624	117.7	12.0	33593	456.9	27866	524.1	7.4	0	7.4	3.38
2003	598	125.5	13.3	37912	276.8	30741	627.8	3.7	0	3.7	2.42
2203	540	124.7	13.9	43364	216.8	32201	668.3	7.0	0	7.0	2.48

Table B8 Brake specific fuel consumption and brake specific emissions in g/kW.h at maximum torque.
Diesel only operation - after conversion (DOC).

Speed rpm	bsfcD g/kWh	bsfcNG g/kWh	bsO2 g/kWh	bsCO g/kWh	bsCO2 g/kWh	bsNOx g/kWh	bsTHC g/kWh	bsCH4 g/kWh	bsRHC g/kWh	bsSoot g/kWh
805	275.1	0	785.1	75.34	739.5	11.74	0.29	0	0.29	0.103
1001	265.2	0	822.9	37.28	741.8	12.78	0.23	0	0.23	0.063
1201	254.8	0	865.5	32.41	763.0	12.31	0.35	0	0.35	0.076
1401	254.9	0	903.0	20.88	789.7	14.94	0.20	0	0.20	0.063
1602	263.8	0	968.5	17.79	820.2	14.58	0.41	0	0.41	0.065
1803	270.0	0	1027.4	13.97	852.2	16.03	0.23	0	0.23	0.103
2003	270.8	0	1087.5	7.94	881.8	18.01	0.11	0	0.11	0.069
2203	284.9	0	1252.3	6.26	929.9	19.30	0.20	0	0.20	0.072

Table B9 Fuel consumption and raw emissions at maximum torque.
Dual fuel operation - tubes pointed downwards (DFdown).

Speed rpm	Tq Nm	NG g/s	Diesel g/s	Air g/s	O2 %	CO %	CO2 %	NOx ppm	THC ppm C6	CH4 ppm C	RHC ppm C6	Tr %
807	712	3.088	2.529	119	11.1	0.761	5.77	925	2525	15966	0	99.18
1001	721	3.069	3.400	153	11.0	0.476	5.86	1027	1923	12061	0	99.48
1201	713	3.573	4.078	186	11.1	0.581	5.81	881	1722	10608	0	99.10
1402	697	3.797	4.748	218	11.0	0.451	6.00	970	1651	10128	0	99.42
1602	652	3.409	5.613	248	11.2	0.253	6.06	897	1199	7412	0	99.62
1803	623	3.258	6.751	280	11.2	0.208	6.13	866	663	6144	0	99.60
2003	624	2.536	8.286	310	10.8	0.186	6.55	1113	446	4033	0	99.63
2203	584	1.767	9.614	341	11.0	0.175	6.54	1126	277	2609	0	99.69

Table B10 Corrected torque, corrected power, air box temperature, tailpipe temperature, A/F ratio, carbon balance, H/C ratio, and thermal efficiency at maximum torque.
Dual fuel operation - tubes pointed downwards (DFdown).

Speed rpm	Tqcor Nm	Powcor kW	BoxTem deg C	TailTem deg C	A/F mass	A/F emm	Cbal in/out %	H/C emm	Eff %
807	726.2	61.4	63.7	360	21.14	24.39	112	2.8	24.2
1001	736.5	77.2	66.5	370	23.72	26.26	110	2.9	26.5
1201	725.8	91.3	68.1	411	24.36	26.68	109	2.8	26.6
1402	708.6	104.0	76.2	389	25.53	26.71	104	2.8	27.2
1602	661.6	111.0	82.6	399	27.53	28.49	103	2.7	27.6
1803	630.6	119.0	89.6	399	28.00	29.05	104	2.7	26.8
2003	630.7	132.3	100.3	438	28.68	28.71	100	2.5	27.7
2203	589.4	136.0	111.0	444	29.99	29.63	99	2.3	27.2

Table B11 Observed torque, observed power, and emissions in mg/s (g/s) at maximum torque.
Dual fuel operation - tubes pointed downwards (DFdown).

Speed rpm	Tq Nm	Pow kW	H2O g/s	O2 mg/s	CO mg/s	CO2 mg/s	NOx mg/s	THC mg/s	CH4 mg/s	RHC mg/s	Soot mg/s
807	712	60.2	5.9	14474	872.5	10389	174.2	994.8	1048.4	0	1.45
1001	721	75.6	7.9	18533	698.9	13518	247.7	970.4	1014.3	0	1.19
1201	713	89.7	9.6	22523	1034.6	16245	257.5	1053.7	1081.9	0	2.46
1402	697	102.3	11.2	26030	937.3	19601	331.0	1179.0	1205.5	0	1.88
1602	652	109.3	12.3	30239	597.4	22463	347.9	972.9	1002.3	0	1.38
1803	623	117.5	13.7	34143	553.4	25630	378.3	605.8	936.3	0	1.66
2003	624	130.8	15.2	36351	546.0	30191	536.3	450.0	678.1	0	1.70
2203	584	134.8	15.8	40735	565.1	33205	597.3	306.9	482.5	0	1.57

Table B12 Brake specific fuel consumption and brake specific emissions in g/kW.h at maximum torque.
Dual fuel operation - tubes pointed downwards (DFdown).

Speed rpm	bsfcD g/kWh	bsfcNG g/kWh	bsO2 g/kWh	bsCO g/kWh	bsCO2 g/kWh	bsNOx g/kWh	bsTHC g/kWh	bsCH4 g/kWh	bsRHC g/kWh	bsSoot g/kWh
807	151.3	184.7	865.9	52.20	621.5	10.42	59.52	62.73	0	0.087
1001	161.9	146.2	882.5	33.28	643.7	11.79	46.21	48.30	0	0.057
1201	163.7	143.4	904.1	41.53	652.0	10.34	42.29	43.43	0	0.099
1402	167.1	133.6	916.0	32.98	689.8	11.65	41.49	42.42	0	0.066
1602	184.8	112.3	995.6	19.67	739.6	11.45	32.03	33.00	0	0.045
1803	206.7	99.8	1045.6	16.95	784.9	11.59	18.55	28.68	0	0.051
2003	228.0	69.8	1000.3	15.03	830.8	14.77	12.38	18.66	0	0.047
2203	256.8	47.2	1088.2	15.10	887.1	15.96	8.20	12.89	0	0.042

Table B13 Fuel consumption and raw emissions at maximum torque.
Dual fuel operation - tubes pointed upwards (DFup).

Speed rpm	Tq Nm	NG g/s	Diesel g/s	Air g/s	O2 %	CO %	CO2 %	NOx ppm	THC ppm C6	CH4 ppm C	RHC ppm C6	Tr %
802	688	3.248	2.467	114	10.6	1.302	5.46	873	2996	17262	119	97.81
1003	712	3.362	3.467	150	10.6	0.902	5.79	896	2528	14650	86	98.64
1204	703	3.565	4.368	181	10.5	1.089	5.69	847	1901	10983	70	97.93
1404	674	4.071	5.272	211	10.0	1.065	6.05	887	1934	11298	51	98.29
1604	646	3.749	5.808	240	10.5	0.582	6.05	892	1539	8889	57	98.76
1805	614	2.972	6.667	268	10.8	0.382	6.18	904	1057	5921	70	98.72
2006	593	1.594	8.494	299	10.8	0.236	6.45	1019	488	2725	34	99.22
2207	523	0.633	9.231	329	11.7	0.111	6.06	1038	144	744	20	99.49

Table B14 Corrected torque, corrected power, air box temperature, tailpipe temperature, A/F ratio, carbon balance, H/C ratio, and thermal efficiency at maximum torque.
Dual fuel operation - tubes pointed upwards (DFup).

Speed rpm	Tqcor Nm	Powcor kW	BoxTem deg C	TailTem deg C	A/F mass	A/F emm	Cbal in/out %	H/C emm	Eff %
802	731.0	61.4	67.2	384	19.88	22.67	111	2.7	22.8
1003	755.4	79.3	68.9	382	21.95	23.78	107	2.7	24.8
1204	739.1	93.2	72.6	417	22.80	24.78	108	2.7	25.3
1404	704.7	103.6	80.9	422	22.61	23.76	104	2.7	24.0
1604	669.9	112.5	89.0	419	25.15	26.02	104	2.8	25.8
1805	637.9	120.6	95.7	414	27.83	27.72	100	2.5	27.5
2006	611.7	128.5	106.4	433	29.68	29.04	99	2.3	28.4
2207	537.5	124.2	113.5	415	33.36	32.74	99	2.1	28.3

Table B15 Observed torque, observed power, and emissions in mg/s (g/s) at maximum torque.
Dual fuel operation - tubes pointed upwards (DFup).

Speed rpm	Tq Nm	Pow kW	H2O g/s	O2 mg/s	CO mg/s	CO2 mg/s	NOx mg/s	THC mg/s	CH4 mg/s	RHC mg/s	Soot mg/s
802	688	57.8	5.9	13535	1454.0	9589	160.2	1149.7	1104.1	40.7	3.69
1003	712	74.8	7.7	17675	1319.6	13304	215.3	1271.0	1227.5	38.8	3.02
1204	703	88.6	9.6	21137	1914.7	15706	244.6	1148.3	1106.0	37.8	5.53
1404	674	99.0	11.7	23432	2180.1	19464	298.2	1360.7	1324.6	32.3	5.35
1604	646	108.5	12.9	27739	1350.4	22042	339.8	1227.1	1181.3	40.9	4.41
1805	614	116.0	13.2	31853	987.4	25107	383.5	938.9	876.5	55.7	5.05
2006	593	124.5	14.1	35417	678.6	29154	481.1	482.3	448.8	29.9	3.44
2207	523	120.9	13.4	42389	351.9	30204	540.5	157.0	135.1	19.6	2.46

Table B16 Brake specific fuel consumption and brake specific emissions in g/kW.h at maximum torque.
Dual fuel operation - tubes pointed upwards (DFup).

Speed rpm	bsfcD g/kWh	bsfcNG g/kWh	bsO2 g/kWh	bsCO g/kWh	bsCO2 g/kWh	bsNOx g/kWh	bsTHC g/kWh	bsCH4 g/kWh	bsRHC g/kWh	bsSoot g/kWh
802	153.7	202.4	843.3	90.59	597.4	9.98	71.63	68.79	2.54	0.23
1003	166.8	161.8	850.4	63.49	640.1	10.36	61.15	59.06	1.87	0.15
1204	177.5	144.8	858.9	77.80	638.2	9.94	46.66	44.94	1.54	0.22
1404	191.6	147.9	851.7	79.24	707.4	10.84	49.46	48.14	1.17	0.19
1604	192.8	124.4	920.5	44.81	731.5	11.28	40.72	39.20	1.36	0.15
1805	206.8	92.2	988.1	30.63	778.9	11.89	29.13	27.19	1.73	0.16
2006	245.7	46.1	1024.3	19.63	843.2	13.92	13.95	12.98	0.87	0.10
2207	274.9	18.9	1262.4	10.48	899.5	16.09	4.68	4.02	0.58	0.07

APPENDIX C

THE JAPANESE 13-MODE TEST RESULTS

Table C1 The 13-mode test fuel consumption and raw emissions.
Diesel only operation - baseline (DOB).

Mode #	Speed rpm	Tq Nm	NG g/s	Diesel g/s	O2 %	CO %	CO2 %	NOx ppm	THC ppm C6	CH4 ppm C	RHC ppm C6	Smk HSU
1	572	7	0	0.453	19.0	0.033	1.12	194	20	0	20	1
2	878	144	0	1.714	17.4	0.032	2.10	401	17	0	17	1
3	878	288	0	2.065	16.1	0.026	2.96	606	14	0	14	2
4	571	7	0	0.442	19.1	0.024	1.08	190	22	0	22	1
5	1318	144	0	2.135	17.2	0.024	2.24	389	18	0	18	1
6	1319	287	0	3.183	15.8	0.024	3.13	576	14	0	14	1
7	1759	256	0	4.223	15.7	0.024	3.22	524	14	0	14	1
8	1759	384	0	5.476	14.3	0.024	4.04	722	14	0	14	1
9	1319	432	0	4.138	14.3	0.024	3.98	786	13	0	13	1
10	1319	573	0	5.273	12.6	0.024	5.16	967	14	0	14	2
11	1319	680	0	6.560	10.8	0.084	6.35	946	19	0	19	6
12	1759	513	0	6.909	12.5	0.024	5.20	900	16	0	16	2
13	1319	37	0	1.400	18.5	0.025	1.49	211	22	0	22	1

Table C2 Weighting factors, power, air flow, tailpipe temperature, A/F ratio, carbon balance, H/C ratio, and thermal efficiency for the 13-mode test. Diesel only operation - baseline (DOB).

Mode #	WF	Pow kW	Air g/s	TailTem deg C	A/F mass	A/F emm	Cbal in/out %	H/C emm	Eff %
1	0.205	0.4	78	71	174.7	165.0	73	3.1	5.2
2	0.037	13.3	137	113	81.5	90.6	122	3.1	18.3
3	0.027	26.5	139	151	67.3	65.3	104	3.0	29.7
4	0.205	0.4	78	72	178.8	171.3	99	3.0	3.0
5	0.029	19.9	211	149	99.0	84.9	93	3.1	21.5
6	0.064	39.6	211	196	66.4	62.1	100	3.0	28.8
7	0.041	47.2	280	236	66.4	60.5	97	3.0	25.9
8	0.032	70.7	279	279	50.9	48.2	102	3.0	29.9
9	0.077	59.7	212	262	51.2	48.7	103	3.1	33.4
10	0.055	79.1	212	305	40.1	38.5	102	2.9	34.8
11	0.049	94.0	210	374	32.1	31.4	104	2.9	33.2
12	0.037	94.4	277	362	40.1	38.1	101	3.0	31.7
13	0.142	5.1	211	155	150.8	127.3	90	2.9	8.5

Table C3 The 13-mode test emissions in mg/s (g/s).
Diesel only operation - baseline (DOB).

Mode #	H2O g/s	O2 mg/s	CO mg/s	CO2 mg/s	NOx mg/s	THC mg/s	CH4 mg/s	RHC mg/s	Soot mg/s
1	0.9	16356	25.2	1324	24.0	4.5	0	4.5	0.278
2	2.8	26241	41.5	4341	86.8	6.7	0	6.7	0.492
3	3.9	24370	34.7	6166	132.1	5.5	0	5.5	1.002
4	0.8	16522	18.4	1292	23.7	5.0	0	5.0	0.279
5	4.6	39800	49.1	7136	129.5	10.9	0	10.9	0.758
6	6.1	36479	48.7	9919	190.9	8.8	0	8.8	0.762
7	8.2	47982	64.6	13520	230.3	11.7	0	11.7	1.011
8	10.5	43185	63.8	16790	313.4	11.2	0	11.2	1.010
9	8.1	32814	48.4	12547	258.9	8.3	0	8.3	1.767
10	9.7	28766	48.1	16152	316.2	8.6	0	8.6	1.541
11	11.7	24179	164.3	19607	305.6	11.3	0	11.3	4.626
12	12.9	37318	63.0	21281	384.9	12.5	0	12.5	2.017
13	2.9	42995	50.2	4786	70.7	13.5	0	13.5	0.754

Table C4 The 13-mode test brake specific fuel consumption and brake specific emissions in g/kW.h.
Diesel only operation - baseline (DOB).

Mode #	bsfcD g/kWh	bsfcNG g/kWh	bsO2 g/kWh	bsCO g/kWh	bsCO2 g/kWh	bsNOx g/kWh	bsTHC g/kWh	bsCH4 g/kWh	bsRHC g/kWh	bsSoot g/kWh
2	464.9	0	7117	11.26	1177.4	23.55	1.83	0	1.83	0.133
3	280.6	0	3311	4.71	837.8	17.95	0.75	0	0.75	0.136
5	387.1	0	7215	8.91	1293.6	23.48	1.98	0	1.98	0.138
6	289.2	0	3315	4.43	901.3	17.34	0.80	0	0.80	0.069
7	322.1	0	3659	4.92	1031.0	17.56	0.90	0	0.90	0.077
8	278.7	0	2198	3.25	854.6	15.95	0.57	0	0.57	0.051
9	249.7	0	1980	2.92	757.2	15.63	0.50	0	0.50	0.046
10	239.9	0	1309	2.19	734.9	14.39	0.39	0	0.39	0.070
11	251.3	0	927	6.30	751.3	11.71	0.44	0	0.44	0.177
12	263.4	0	1423	2.40	811.3	14.68	0.48	0	0.48	0.077
13	981.7	0	30147	35.22	3355.8	49.59	0.43	0	0.43	0.529

Table C5 The 13-mode test fuel consumption and raw emissions.
Diesel only operation - after conversion (DOC).

Mode #	Speed rpm	Tq Nm	NG g/s	Diesel g/s	O2 %	CO %	CO2 %	NOx ppm	THC ppm C6	CH4 ppm C	RHC ppm C6	Tr %
1	575	6	0	0.256	18.7	0.016	1.04	153	20	0	20	99.90
2	877	143	0	1.328	17.6	0.014	2.04	274	10	0	10	99.90
3	880	287	0	1.967	16.3	0.007	2.93	487	9	0	9	99.90
4	579	6	0	0.329	18.9	0.015	1.05	134	22	0	22	99.86
5	1320	147	0	2.168	17.2	0.011	2.26	284	9	0	9	99.81
6	1320	284	0	3.109	16.0	0.005	3.08	529	8	0	8	99.78
7	1761	256	0	4.414	15.9	0.005	3.21	439	7	0	7	99.83
8	1762	385	0	5.559	14.6	0.005	4.09	788	7	0	7	99.86
9	1322	424	0	4.133	14.6	0.002	3.98	773	8	0	8	99.80
10	1322	570	0	5.272	13.0	0.015	5.24	1123	10	0	10	99.79
11	1322	675	0	6.448	11.5	0.172	6.33	1292	15	0	15	99.58
12	1762	515	0	6.595	13.0	0.017	5.26	1058	12	0	12	99.80
13	1321	29	0	1.361	18.3	0.019	1.45	177	17	0	17	99.77

Table C6 Weighting factors, power, air flow, air box temperature, tailpipe temperature, A/F ratio, carbon balance, H/C ratio, and thermal efficiency for the 13-mode test.
Diesel only operation - after conversion (DOC).

Mode #	WF	Pow kW	Air g/s	BoxTem deg C	TailTem deg C	A/F mass	A/F emm	Cbal in/out %	H/C emm	Eff %
1	0.205	0.4	67	55.7	77	406.8	160.8	73	4.9	5.2
2	0.037	13.2	137	55.5	114	100.9	94.8	102	2.9	23.4
3	0.027	26.5	132	56.1	164	67.2	67.3	105	2.8	31.2
4	0.205	0.4	64	54.0	67	215.8	165.4	99	4.3	3.0
5	0.029	20.3	207	58.3	141	95.6	85.6	95	2.9	21.7
6	0.064	39.2	207	61.6	191	66.6	64.3	102	2.8	29.2
7	0.041	47.2	279	71.4	217	63.2	62.0	102	2.6	24.8
8	0.032	71.0	277	76.8	267	49.8	49.1	103	2.6	29.6
9	0.077	58.7	208	65.7	243	50.3	50.0	105	2.8	32.9
10	0.055	78.9	208	68.9	302	39.4	38.9	102	2.5	34.7
11	0.049	93.4	206	73.0	379	31.9	32.1	102	2.2	33.6
12	0.037	94.9	275	83.1	327	41.8	38.8	96	2.4	33.4
13	0.142	3.9	208	59.7	108	153.1	124.6	92	3.8	6.7

Table C7 The 13-mode test emissions in mg/s (g/s).
Diesel only operation - after conversion (DOC).

Mode #	H2O g/s	O2 mg/s	CO mg/s	CO2 mg/s	NOx mg/s	THC mg/s	CH4 mg/s	RHC mg/s	Soot mg/s
1	1.1	13896	10.4	1064	16.3	4.0	0	4.0	0.093
2	2.4	25438	17.7	4053	57.1	3.9	0	3.9	0.191
3	3.3	25577	8.9	5843	101.5	3.4	0	3.4	0.197
4	0.9	13321	9.3	1019	13.6	4.3	0	4.3	0.132
5	4.3	39268	22.0	7101	93.2	5.5	0	5.5	0.553
6	5.4	36341	9.9	9585	172.4	4.8	0	4.8	0.645
7	7.3	48587	13.4	13514	193.1	5.9	0	5.9	0.699
8	9.1	43954	13.2	16967	342.0	5.6	0	5.6	0.580
9	7.0	33103	4.0	12382	251.8	4.9	0	4.9	0.594
10	8.2	29171	29.5	16204	363.2	6.0	0	6.0	0.641
11	8.9	25639	335.4	19382	413.6	8.9	0	8.9	1.257
12	10.7	38705	44.4	21587	453.5	9.9	0	9.9	0.814
13	3.6	41874	38.2	4573	58.5	10.6	0	10.6	0.673

Table C8 The 13-mode test brake specific fuel consumption and brake specific emissions in g/kW.h.
Diesel only operation - after conversion (DOC).

Mode #	bsfcD g/kWh	bsfcNG g/kWh	bsO2 g/kWh	bsCO g/kWh	bsCO2 g/kWh	bsNOx g/kWh	bsTHC g/kWh	bsCH4 g/kWh	bsRHC g/kWh	bsSoot g/kWh
2	361.8	0	6950	4.85	1107.3	15.60	1.05	0	1.05	0.052
3	267.5	0	3206	1.21	794.4	13.80	0.47	0	0.47	0.027
5	385.1	0	6974	3.90	1261.2	16.55	0.98	0	0.98	0.098
6	285.5	0	3337	0.91	880.2	15.83	0.44	0	0.44	0.059
7	336.9	0	3708	1.02	1031.4	14.74	0.45	0	0.45	0.053
8	281.7	0	2227	0.67	859.8	17.33	0.28	0	0.28	0.029
9	253.7	0	2032	0.24	760.0	15.45	0.30	0	0.30	0.036
10	240.5	0	1331	1.35	739.1	16.57	0.27	0	0.27	0.029
11	248.5	0	988	12.92	746.9	15.94	0.34	0	0.34	0.048
12	250.0	0	1468	1.68	818.5	17.20	0.37	0	0.37	0.031
13	1241.7	0	38175	34.79	4196.1	53.29	9.66	0	9.66	0.614

Table C9 The 13-mode test fuel consumption and raw emissions. Dual fuel operation - tubes pointed downwards (DFdown).

Mode #	Speed rpm	Tq Nm	NG g/s	Diesel g/s	O2 %	CO %	CO2 %	NOx ppm	THC ppm C6	CH4 ppm C	RHC ppm C6	Tr %
1	575	6	0	0.256	18.7	0.016	1.04	153	20	0	20	99.90
2	875	138	0.284	0.877	17.6	0.035	1.95	181	247	1385	17	99.85
3	879	285	0.913	1.680	16.2	0.055	2.73	404	865	4945	41	99.83
4	579	6	0	0.329	18.9	0.015	1.05	134	22	0	22	99.86
5	1320	147	0.499	1.830	17.3	0.033	2.14	163	322	1778	26	99.73
6	1320	282	1.373	2.406	15.9	0.053	2.91	246	803	4549	45	99.75
7	1762	250	1.804	3.479	15.7	0.055	3.07	187	814	4645	40	99.80
8	1762	382	2.564	4.076	14.3	0.072	3.86	498	989	5770	28	99.76
9	1321	430	3.060	2.714	14.2	0.099	3.72	515	1584	9505	5	99.76
10	1321	571	3.289	3510	12.8	0.181	4.66	678	1556	9403	0	99.66
11	1322	675	3.451	4.492	11.3	0.463	5.58	932	1572	9644	0	99.08
12	1763	516	2.921	4.787	12.8	0.101	4.86	690	1009	6039	2	99.61
13	1321	29	0	1.361	18.3	0.019	1.45	177	17	0	17	99.77

Table C10 Weighting factors, power, air flow, air box temperature, tailpipe temperature, A/F ratio, carbon balance, H/C ratio, and thermal efficiency for the 13-mode test.
Dual fuel operation - tubes pointed downwards (DFdown).

Mode #	WF	Pow kW	Air g/s	BoxTem deg C	TailTem deg C	A/F mass	A/F emm	Cbal in/out %	H/C emm	Eff %
1	0.205	0.4	67	55.7	77	406.8	160.8	73	4.9	5.2
2	0.037	12.6	130	55.3	107	112.2	89.7	84	3.1	25.0
3	0.027	26.2	131	56.5	149	50.6	58.0	119	3.2	23.0
4	0.205	0.4	64	54.0	67	215.8	165.4	99	4.3	3.0
5	0.029	19.5	207	59.1	132	88.9	81.0	96	3.2	19.2
6	0.064	39.0	207	61.8	173	54.8	55.5	105	3.2	23.5
7	0.041	46.2	278	73.1	203	52.6	53.3	105	3.2	19.9
8	0.032	70.5	276	78.2	264	41.6	42.5	106	3.3	24.2
9	0.077	59.5	208	65.0	238	35.9	39.5	113	3.5	23.3
10	0.055	79.0	207	67.4	317	30.4	33.3	112	3.3	26.3
11	0.049	93.5	205	71.5	377	25.8	28.1	110	3.0	26.7
12	0.037	95.2	275	83.5	335	35.6	35.2	101	3.1	28.1
13	0.142	3.9	208	59.7	108	153.0	124.6	92	3.8	6.7

Table C11 The 13-mode test emissions in mg/s (g/s).
Dual fuel operation - tubes pointed downwards (DFdown).

Mode #	H2O g/s	O2 mg/s	CO mg/s	CO2 mg/s	NOx mg/s	THC mg/s	CH4 mg/s	RHC mg/s	Soot mg/s
1	1.1	13896	10.4	1064	16.3	4.0	0	4.0	0.093
2	2.5	25199	43.9	3849	37.2	106.7	99.5	6.4	0.288
3	3.5	23485	69.7	5424	84.1	376.3	358.7	15.8	0.315
4	0.9	13327	9.3	1019	13.6	4.3	0	4.3	0.132
5	4.3	39371	65.8	6713	53.4	220.7	203.1	15.7	0.796
6	5.9	36222	105.5	9113	80.5	549.1	518.6	27.3	0.758
7	8.2	48037	147.1	12896	81.9	474.9	711.4	32.6	0.808
8	10.5	43157	190.1	16026	215.8	897.5	872.4	22.5	0.943
9	8.4	32405	197.3	11654	168.4	1084.7	1084.9	2.9	0.737
10	9.7	28892	357.6	14466	220.1	1056.8	1064.1	0	1.021
11	10.8	25288	905.0	17135	299.1	1055.7	1079.7	0	2.766
12	12.3	38276	263.9	19968	296.1	905.4	903.6	1.8	1.549
13	3.6	41874	38.2	4573	58.5	10.6	0	10.6	0.673

Table C12 The 13-mode test brake specific fuel consumption and brake specific emissions in g/kW.h.
Dual fuel operation - tubes pointed downwards (DFdown).

Mode #	bsfcD g/kWh	bsfcNG g/kWh	bsO2 g/kWh	bsCO g/kWh	bsCO2 g/kWh	bsNOx g/kWh	bsTHC g/kWh	bsCH4 g/kWh	bsRHC g/kWh	bsSoot g/kWh
2	249.8	80.8	7175	12.50	1095.9	10.60	30.37	28.33	1.83	0.082
3	230.8	125.4	3226	9.57	745.1	11.55	51.69	49.27	2.16	0.043
5	337.7	92.1	7264	12.14	1238.6	9.86	40.72	37.48	2.90	0.147
6	222.3	126.8	3347	9.75	842.0	11.44	50.74	47.92	2.52	0.070
7	271.2	140.6	3744	11.46	1005.0	6.38	58.29	55.44	2.54	0.063
8	208.2	131.0	2205	9.71	818.7	11.02	45.85	44.57	1.15	0.048
9	164.2	185.1	1960	11.94	705.0	10.19	65.62	65.43	0.17	0.045
10	160.0	149.9	1317	16.31	659.6	10.04	48.18	48.52	0	0.047
11	173.0	132.9	974	34.85	659.8	11.52	40.65	41.58	0	0.107
12	181.0	110.5	1448	9.98	755.3	11.20	34.25	34.18	0.07	0.059
13	1241.7	0	38175	34.79	4169.1	53.29	9.66	0	9.66	0.614

Table C13 The 13-mode fuel consumption and raw emissions. Dual fuel operation - tubes pointed upwards (DFup).

Mode #	Speed rpm	Tq Nm	NG g/s	Diesel g/s	O2 %	CO %	CO2 %	NOx ppm	THC ppm C6	CH4 ppm C	RHC ppm C6	Tr %
1	577	9	0	0.486	19.3	0.013	1.15	176	21	0	21	99.77
2	882	145	0.375	1.109	17.7	0.048	2.06	309	397	2027	59	99.65
3	882	289	0.907	1.476	16.3	0.054	2.86	478	811	4211	110	99.63
4	578	9	0	0.274	19.2	0.018	1.15	181	42	0	42	99.66
5	1323	144	0.396	1.803	17.4	0.035	2.26	312	249	837	109	99.58
6	1323	285	1.278	2.436	16.0	0.054	3.09	462	794	4225	90	99.56
7	1765	255	1.521	3.378	15.8	0.049	3.22	408	734	3893	85	99.61
8	1766	386	2.978	3.804	14.2	0.101	3.99	535	1250	6696	134	99.44
9	1324	427	2.882	2.554	14.1	0.154	3.89	618	1591	8773	128	99.50
10	1323	575	3.438	3.523	12.4	0.422	4.75	788	1647	9522	60	98.81
11	1324	675	3.723	4.648	10.8	0.999	5.61	848	1734	10476	0	97.60
12	1765	503	3.116	4.945	12.8	0.118	4.91	746	1138	6622	35	99.52
13	1324	29	0	1.453	18.7	0.024	1.54	177	177	0	29	99.74

Table C14 Weighting factors, power, air flow, air box temperature, tailpipe temperature, A/F ratio, carbon balance, H/C ratio, and thermal efficiency for the 13-mode test.
Dual fuel operation - tubes pointed upwards (DFup).

Mode #	WF	Pow kW	Air g/s	BoxTem deg C	TailTem deg C	A/F mass	A/F emm	Cbal in/out %	H/C emm	Eff %
1	0.205	0.5	73	62.7	65	165.6	194.1	115	0.4	2.7
2	0.037	13.4	130	60.6	110	87.5	86.9	96	1.9	20.7
3	0.027	26.7	131	61.5	145	55.1	58.2	104	2.4	25.6
4	0.205	0.5	70	60.2	70	316.2	187.6	66	0.6	5.4
5	0.029	20.0	202	63.8	135	92.3	83.9	89	1.9	21.0
6	0.064	39.5	202	66.0	175	54.5	54.9	99	2.3	24.3
7	0.041	47.1	269	76.8	208	54.9	53.5	96	2.4	21.9
8	0.032	71.3	266	83.0	265	39.3	40.1	102	2.8	23.9
9	0.077	59.2	202	69.9	236	37.1	38.3	103	3.0	24.6
10	0.055	79.7	202	71.8	324	29.0	31.2	108	3.0	25.9
11	0.049	93.6	200	66.1	397	23.9	25.6	106	2.7	25.4
12	0.037	93.0	265	73.5	341	32.8	34.3	105	2.8	26.2
13	0.142	4.0	202	64.6	115	139.4	138.1	92	0.9	6.4

Table C15 The 13-mode test emissions in mg/s (g/s).
Dual fuel operation - tubes pointed upwards (DFup).

Mode #	H2O g/s	O2 mg/s	CO mg/s	CO2 mg/s	NOx mg/s	THC mg/s	CH4 mg/s	RHC mg/s	Soot mg/s
1	0.1	15986	9.4	1311	20.9	5.2	0	5.2	0.237
2	1.5	25847	61.5	4139	65.1	174.8	148.7	23.3	0.651
3	2.6	23968	69.6	5785	101.2	359.6	311.0	43.4	0.705
4	0.1	15315	12.6	1260	20.8	10.2	0	10.2	0.345
5	2.6	39542	69.6	7055	101.9	170.1	95.3	66.8	1.215
6	4.3	36227	107.2	9630	150.6	541.8	480.5	54.8	1.291
7	6.0	47384	129.0	13331	176.5	663.7	586.9	68.6	1.505
8	8.8	42136	262.1	16263	227.9	1114.7	995.2	106.7	2.153
9	7.2	31713	303.4	12041	200.0	1076.9	989.9	77.7	1.483
10	9.1	27813	827.6	14652	253.8	1109.9	1069.6	36.1	3.529
11	10.3	23904	1938.9	17119	270.2	1156.6	1164.5	0	7.080
12	10.8	37459	302.7	19782	314.3	1003.5	972.9	27.3	1.856
13	0.9	42732	48.1	4843	58.4	20.0	0	20.0	0.764

Table C16 The 13-mode test brake specific fuel consumption and brake specific emissions in g/kW.h.
Dual fuel operation - tubes pointed upwards (DFup).

Mode #	bsfcD g/kWh	bsfcNG g/kWh	bsO2 g/kWh	bsCO g/kWh	bsCO2 g/kWh	bsNOx g/kWh	bsTHC g/kWh	bsCH4 g/kWh	bsRHC g/kWh	bsSoot g/kWh
2	297.9	101.0	6946	16.52	1112.2	17.49	46.98	39.96	6.27	0.175
3	198.8	122.2	3228	9.38	779.1	13.62	48.43	41.89	5.85	0.095
5	324.7	71.3	7117	12.52	1269.7	18.33	30.61	17.15	12.02	0.219
6	221.9	116.3	3299	9.76	877.0	13.72	49.34	43.75	4.99	0.118
7	258.3	116.3	3623	9.86	1019.3	13.50	50.75	44.88	5.25	0.115
8	192.0	150.3	2127	13.23	820.8	11.50	56.25	50.23	5.39	0.109
9	155.3	175.2	1928	18.45	732.2	12.16	65.48	60.20	4.72	0.090
10	159.1	155.2	1256	37.36	661.4	11.46	50.10	48.28	1.63	0.159
11	178.7	143.1	919	74.55	658.2	10.39	44.47	44.77	0	0.272
12	191.5	120.6	1450	11.72	765.7	12.17	38.84	37.66	1.06	0.072
13	1303.4	0	38314	43.10	4342.3	52.33	17.95	0	17.95	0.685

Table C17 Composite emissions and fuel consumption results for the Japanese 13-mode test.

Emissions and Fuel Consumption	Diesel Only Baseline (g/kW.h)	Diesel Only Converted (g/kW.h)	Dual Fuel Tubes down (g/kW.h)	Dual Fuel Tubes up (g/kW.h)
CO_2	963.39	948.25	889.56	919.36
CO	6.06	4.48	16.91	29.18
NO_x	17.50	17.65	11.81	14.14
THC	1.105	0.802	47.55	49.62
NMHC	1.105	0.802	1.158	3.838
CH_4	0	0	46.56	45.37
SOOT	0.116	0.059	0.083	0.170
Diesel Fuel Consumption	313.17	303.22	230.15	234.42
Natural Gas Consumption	0	0	136.02	136.62